ELEMENTARY
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BY

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PREFACE

The purpose of this book is to present in a clear and concise manner a discussion of the fundamental principles underlying the construction and operation of steam power plant equipment.

The book is the result of the author's experiences while teaching courses in steam engineering at several engineering colleges. The material presented has been used successfully in classroom work, and although intended primarily for men of college grade, it is hoped that it will prove satisfactory for vocational schools and for the general reader seeking information about steam power plants. With this end in view, the mathematical discussion has been kept at a minimum.

The arrangement of the material differs from that of other books in the same field in that an effort has been made to introduce the practical phases of the subject previous to the theoretical. This has been done by having a description of the apparatus precede the theory. This plan has been followed throughout the book with the exception of the chapter on stokers, where it seemed preferable to place the discussion of combustion first. The use of the more important equations is illustrated by examples, and in addition a number of review questions and problems are placed at the end of each chapter in order to give the reader a ready means of testing his knowledge of the material contained in the chapter.

The first chapter discusses the relationship existing between the various equipment used in common types of steam power plants. The remaining chapters naturally divide themselves into two parts. The first part deals with representative types of steam boilers and their auxiliaries. Included in this part is a discussion of physical units; properties of steam; fuels; combustion; methods of burning coal, oil and pulverized fuel; smoke prevention; feedwater purification, and boiler testing. The second part takes up the various types of steam engines, steam turbines, pumps, and condensers. Included in this part is a description of the method used to set the valves of engines; a discussion on the testing of engines and turbines, and a description of the methods used to cool the circulating water for condensers. The last chapter contains a brief description of the equipment used in several modern steam power plants.

The author takes this opportunity to thank and acknowledge his indebtedness to the various manufacturers who so willingly supplied bulletins and illustrations, in many cases at considerable expense to themselves; and regrets that because of the large number they cannot be listed here.

The author also thanks the following for assistance rendered during the preparation of the text: Dean G. C. Anthony; Professors A. C. Willard; A. P. Kratz; Wm. H. Severns; C. H. Chase; F. E. Seavey; J. A. Polson and R. U. Fittz.

Special credit is due the following for permission to use certain material from publications of which they are the authors: Professors J. A. Moyer; C. F. Hirshfeld; C. F. Gebhardt; C. H. Fessenden; E. M. Shealy; Mr. J. F. Cosgrove; F. R. Low, and for permission to use portions of the Power Test Code, The American Society of Mechanical Engineers.

Suggestions or criticisms that will make the book more useful, or for corrections that will make the book more accurate, will be appreciated by the author.

E. MacN.

Tufts College, Mass.,
July, 1923.

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Elementary Steam Power Engineering

CHAPTER I

ELEMENTARY STEAM POWER PLANTS

1. Foreword. — Modern power plants, furnishing mechanical and electrical energy for commercial purposes, use water, gas, oil, or steam to develop power.

The factors which generally determine the type of plant to be used are:

1. Kind of service.
2. Location with regard to fuel and water.
3. Space available.
4. Reliability in operation.
5. Cost to produce a commercial unit of power, when all factors entering into the cost are considered.

The **water power,** or **hydro-electric, plant** may produce a unit of power during a part of the year at a lower cost than a steam power plant of the same capacity, but the water supply may vary sufficiently to lower its capacity during the remainder of the year and thus make necessary an auxiliary source of power. Furthermore, on account of the higher initial cost of the water power installation and the long distributing lines, the cost of power, which would otherwise be low, is increased. The **gas** or **oil power plant** has a high **thermal efficiency,** *or ability to convert the heat supplied into work,* but depreciation, repairs and high cost of fuel generally offset this advantage. The **steam power plant,** because of its dependability and all-round efficiency, can produce power in large quantities at the lowest cost and *is used to supply* **nearly three-quarters** *of the total power now used* in manufacturing, heating, lighting, and railway service in the United States.

2. Classification. — Modern power plants may be classified:

1. According to kind of service $\left\{\begin{array}{l}\text{Stationary}\\ \text{Marine}\\ \text{Locomotive}\\ \text{Automotive}\end{array}\right.$

2. According to the form in which energy is supplied to the main power units	Steam Water Gas Oil
3. According to location with reference to distribution of output	Isolated Central Station
4. According to method of operating main power units	Engine-driven Turbine-driven

Steam power plants are classified as **condensing** or **non-condensing** according to the method used to dispose of the steam exhausted from the main power units. *In the non-condensing plant, exhaust steam from the engine or turbine is discharged at or near the pressure of the atmosphere.* The exhaust steam may be wasted to the air, or it may be used for commercial purposes. *In the condensing plant, the exhaust steam may be discharged into a* **surface condenser,** a vessel closed to the atmosphere and containing a large number of small tubes through which cooling water is forced to circulate, condensing the steam which surrounds the tubes, and lowering the pressure in the vessel; *or the exhaust steam may be discharged into a* **jet condenser,** a closed vessel in which the steam is condensed by coming into direct contact with jets of cooling water which are sprayed into the chamber. The condensed steam is known as **condensate.**

Each type of steam power plant has its field of usefulness. The non-condensing plant is generally used in hotels, office buildings and factories; or in power plants where the exhaust steam can be used for heating, drying, or tanning, or in other manufacturing processes.

Condensing equipment is used in large power plants, such as central stations, where the amount of exhaust steam is in excess of that required for heating and where the maximum amount of power obtainable is desired. Turbine-driven plants are usually operated "condensing," to increase the economy of the turbines.

3. Types of Power Plants to be considered. — The following types of steam power plants will be briefly described, and the arrangement of the apparatus will be shown in outline form to make clear the relation existing between parts of the equipment:

1. Non-condensing isolated plant.
2. Condensing isolated plant.
3. Central station having complete equipment for economical operation.
4. Locomotive power plant.
5. Marine power plant.

4. Essential Steam Power Plant Equipment. — Any steam power plant must have a **furnace** in which fuel is burned to generate heat; a **boiler,** or steel vessel containing water, to utilize the heat generated in the fur-

nace and convert the water into steam; an **engine** or **turbine,** called the **prime mover** or **main unit,** to use the heat energy stored in the steam and perform work; and suitable **piping** to convey the steam. Besides the above equipment, the plant requires numerous **auxiliaries** and **accessories,** which depend upon the location of the plant, the fuel and water available, the service for which the plant is intended, and the economy desired.

5. Non-condensing Isolated Steam Power Plant. — *An isolated station is a power plant that furnishes light, heat or power to a single building or group of buildings situated near each other.* The arrangement of the apparatus in a typical non-condensing isolated plant is shown in Fig. 1. The boiler is a **fire-tube** boiler; that is, the hot gases from the furnace pass through tubes which are surrounded by water. The type of boiler shown is commonly known as a **horizontal return tubular boiler.** It is located within the brick walls of the **boiler setting,** which forms the side and end walls, of the space around the boiler, makes a passage through which the gases from the furnace pass, and serves to confine the heat to the boiler. The brick setting often forms a support for the boiler, but in the best modern practice boilers are suspended from steel I-beams supported by steel columns.

Coal is delivered to the front of the boiler by a car pushed along a track by hand, and is shoveled upon a **stationary grate** under the front half of the boiler. The ashes resulting from burning the coal fall to the **ash-pit** below the grate, and the smoke and hot gases from the furnace pass to the rear of the setting. They then pass through **tubes** to the **smoke connection** at the front of the boiler, and while passing through the tubes give up some of their heat to the water, which surrounds the tubes. From the smoke connection at the front of the boiler, the gases pass, through a circular or rectangular **sheet-metal breeching,** to the **chimney** or **stack,*** which produces sufficient **draft,** or difference of pressure above and below the fire, to supply the air necessary to burn the fuel. A **damper** is placed in the smoke connection to the breeching, to regulate the draft as required to meet the demand for power.

The steam formed in the boiler passes through a **steam nozzle** and **steam pipe** to a **steam header,** from which the **steam lead** to the engine is taken. Part of the steam flowing through the pipe is condensed and carried along by the rapidly moving steam. To prevent this water from entering the cylinder of the engine and causing damage, a **separator** is placed in the steam pipe line, near the engine cylinder, to remove the surplus water from the steam. The water separated from the steam is drained from the separator through a **steam trap** which automatically removes water from a pipe system with minimum loss of steam.

* The word chimney is generally used to denote brick and concrete construction, and stack to denote steel construction.

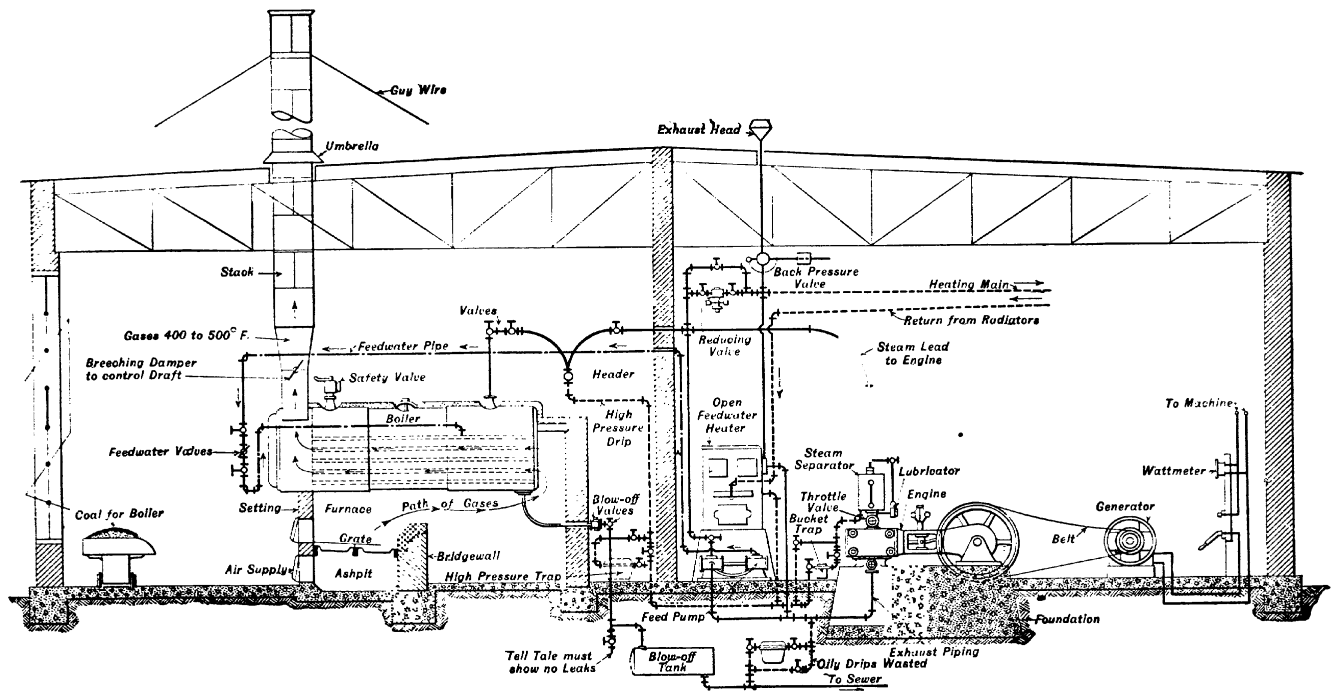


Fig. 1.—Non-condensing Isolated Steam Power Plant.

The steam is used in a **reciprocating engine,** which drives an **electric generator,** by means of a belt, and furnishes current for lighting and other useful purposes. The steam exhausted from the engine cylinder passes through the **exhaust piping** either to an **open feedwater heater** where it comes into direct contact with the feedwater and raises its temperature, or directly to the outside air through a **back-pressure valve** which opens to relieve excessive pressure in the exhaust pipe system. At the top of the exhaust pipe and above the roof of the building, an **exhaust head** is located. It removes oil from the escaping steam and prevents the oil from being thrown over the roof and the surrounding buildings.

To remove the oil carried from the engine cylinder by the exhaust steam, an **oil separator,** which is often a component part of the feedwater heater, is located in the exhaust pipe line just before it enters the heater.

The **boiler feed-pump** takes feedwater from the feedwater heater and forces it into the boiler. The heater is located above the pump and supplies water to the pump under a small pressure, because a pump will not lift hot water through any appreciable distance. The pump is direct-acting, using steam from the main steam line. Exhaust steam from the pump is discharged into the exhaust pipe running to the feedwater heater. The water-end of the pump discharges the feedwater into the boiler through a pipe line running to the front of the boiler and entering the boiler at that end.

During the winter months, exhaust steam from the engine and pump is discharged into the heating system. The back-pressure valve is then set to open at a pressure above that required for heating. As the oil which is picked up by the steam in the engine cylinders is detrimental to the heating system, it is removed before entering the piping to the radiators. In case the amount of exhaust steam from the engines is insufficient to supply the heating requirements, live steam from the main steam line can be admitted to the heating system through a **reducing valve,** which lowers the pressure of the steam from that of the boiler to that required for heating.

Steam passes to the radiators from the exhaust piping near the feedwater heater, and, after being condensed in the radiators or heating coils, is discharged through thermostatic valves into a pipe line which discharges into the feedwater heater or the **returns tank.** Thence it is pumped to the boiler as feedwater.

To operate the equipment shown in Fig. 1, certain **accessories, trimmings, or fittings** are necessary. The boiler requires the following accessories: a **safety valve** to limit the pressure which the boiler may carry; a **water column** and a **water gage** to show the water level in the boiler; a **steam gage** to indicate the pressure of the steam in the boiler; a **blow-off pipe** and valves to discharge the sediment which collects in the boiler

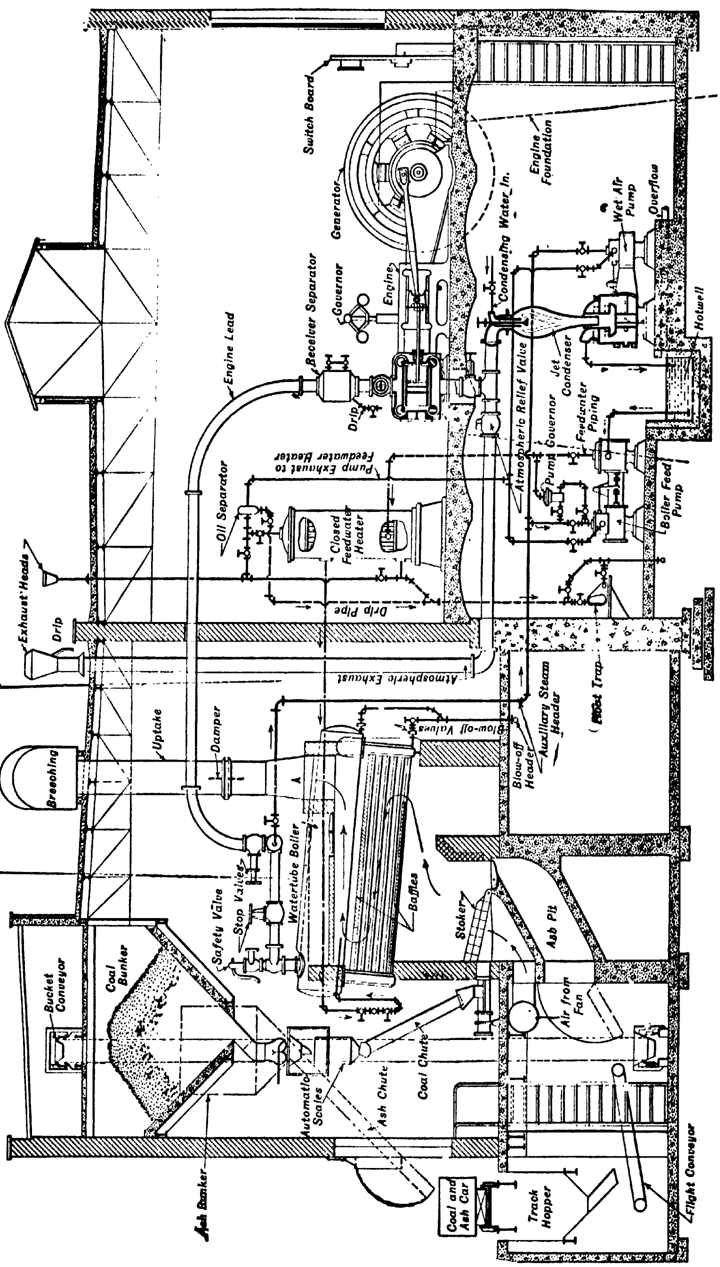


FIG. 2. — Isolated Condensing Steam Power Plant.

shell and to empty the boiler; and a **check valve** in the feedwater line to prevent the return of water from the boiler in case the boiler feed-pump is not working.

The steam unit requires a **throttle valve** to control the flow of steam, **oilers** to supply lubrication to the sliding and rotating parts, and **piping and valves** to drain the condensed steam from the cylinders. As a part of the engine, **a flywheel** is required to prevent excessive fluctuation in speed, and a **governor** to regulate the speed.

6. Condensing Isolated Steam Power Plant. — The apparatus required for the operation of a simple, isolated, condensing power plant is shown in Fig. 2. The boiler is a **water-tube boiler;** that is, it has a series of tubes containing water, around which gases from the furnace pass on their way to the stack. **Baffle walls** force the gases to take the path shown by arrows.

Coal is delivered to a **track hopper** directly from a coal car, and is then discharged upon a **flight conveyor** for delivery to a **bucket conveyor,** which elevates the coal and discharges it into an **overhead bunker.** From the overhead bunker, the coal passes to **automatic scales** and then through a **chute** into the hopper of a stoker having a steam-operated plunger to feed the coal into the furnace. The ashes collect on a **dump plate** at the rear of the furnace and are dumped into the ashpit under the stoker. From the ashpit the ashes are delivered into a bucket conveyor and are elevated to an overhead **ash storage bin.**

Steam is used in a reciprocating engine which is directly connected to an electric generator. Exhaust steam from the engine passes directly into a jet condenser. The condensed steam, together with the condensing water, is removed from the lower part of the condenser by a **steam-driven pump** known as a **vacuum pump,** a **wet-vacuum pump,** or a **wet-air pump** because it removes a mixture of air and water from the condenser, and thus maintains the vacuum, or pressure, below the pressure of the atmosphere. In case it becomes necessary to stop the wet-vacuum pump, provision is made for automatically discharging the exhaust steam to the atmosphere through an **atmospheric relief valve.** The vacuum pump is often a pump which removes air only. It is then called a **dry-vacuum pump,** or a **dry-air pump.**

The exhaust steam from the steam pumps is discharged into a **closed feedwater heater,** in which the steam does not come into direct contact with the feedwater. This heater is generally so constructed that the steam surrounds tubes in which the feedwater is circulating. The remaining equipment is similar to that described under the non-condensing plant.

7. Comparison of Non-condensing and Condensing Plants. — The thermal efficiency of a non-condensing plant, like that shown in Fig. 1,

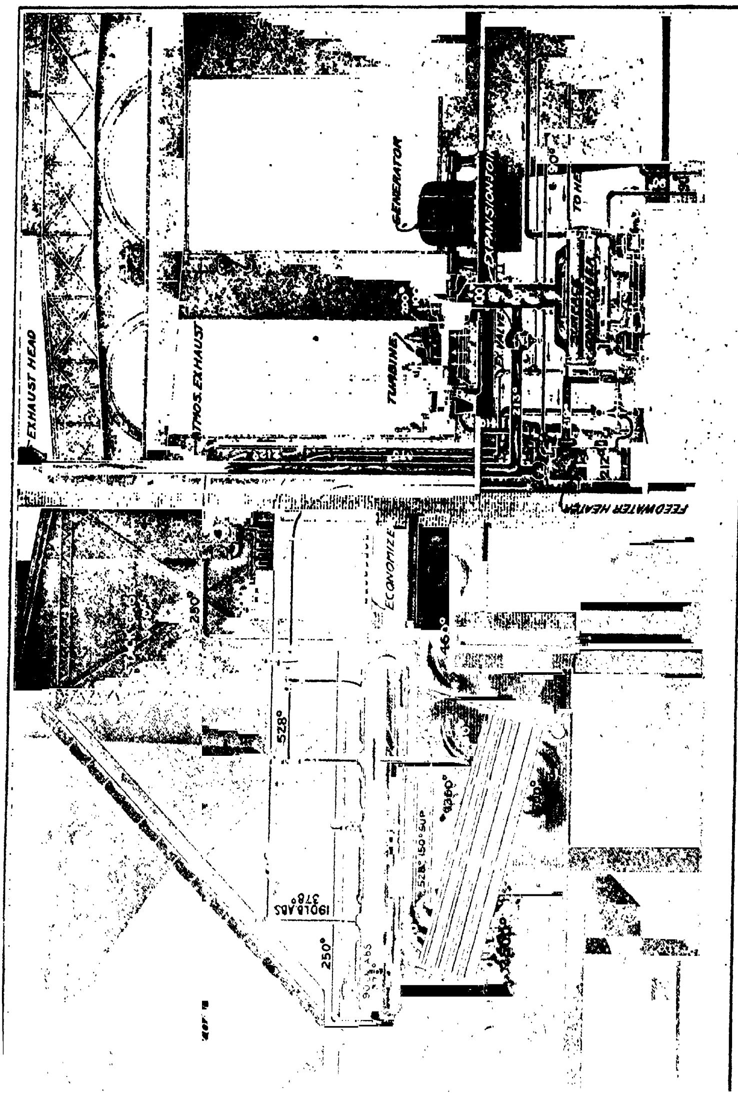


FIG. 3.—Central Station Steam Power Plant.

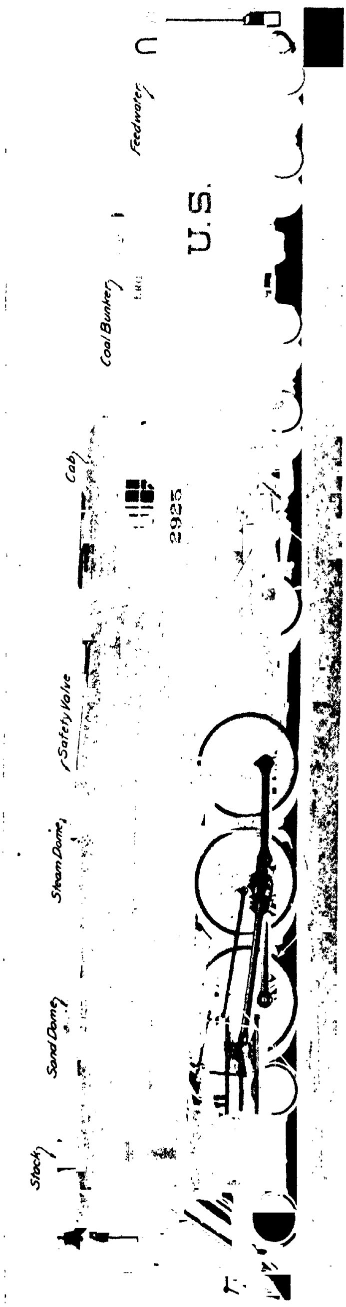
is low. Ordinarily it does not convert more than 4 per cent of the heat energy stored in the coal into work. The thermal efficiency is increased to 6 or 8 per cent during the winter when the exhaust steam is used for heating purposes. A condensing plant like that shown in Fig. 2 is capable of converting 10 per cent of the heat value of the coal into work, and in exceptional cases may run as high as 12 per cent.

In the non-condensing and condensing plants shown in Figs. 1 and 2, no effort was made to bring about all the saving possible by use of special heat-saving equipment. There are several kinds of heat-saving equipment, the installation of which would not be advisable in small power plants, but which might bring about large savings when installed in large central stations.

8. Central Station Steam Power Plant. — *The central station plant is a power plant that supplies power to consumers more or less widely scattered and often far distant from the station.* The layout of such a plant, fully equipped with heat-saving apparatus, is shown in Fig. 3. Coal from the coal bunker is fed into the hopper of a chain grate stoker having a **traveling grate** thus assuring a uniform feeding of coal and a saving in heat. The ashes are discharged from the rear end of the traveling grate into the ashpit, from which they are raked into an **ash car** and moved to the ash storage.

Steam, leaving the boiler, passes through a **superheater,** which consists of a series of coils of pipe connected at each end by a header and located in the path of the hot furnace gases. The gases, having a temperature higher than that of the steam, transmit some of their heat to the steam and thus its temperature is raised above the temperature existing in the boiler. When in this condition, steam is called **superheated steam.**

Steam from the superheater flows through suitable piping to a steam turbine, where a part of its energy is converted into work, and the remainder is exhausted from the turbine into a surface condenser. The condensed steam is pumped from the lower part of the condenser, by a wet-vacuum pump, into a tank or well, called the hot well. The cooling water for the condenser is circulated through the tubes by a piston pump driven by the same steam engine that drives the wet-vacuum pump, and passes the length of the condenser twice. Water at 90 deg. fahr. is pumped from the hot well into an open feedwater heater by a **reciprocating steam-driven pump.** In the feedwater heater it is mixed with the exhaust steam from the steam-driven auxiliaries, such as the boiler feed-pumps and vacuum pumps, and its temperature is raised to about 212 deg. fahr. Leaving the heater at the bottom, the feedwater is pumped by a **centrifugal pump** through an **economizer** into the boiler. The centrifugal pump has a rotating member, or **impeller,** which receives the water near the shaft and discharges it at the periphery of the impeller blades. The econo-



Courtesy American Locomotive Co.

FIG. 4. — Locomotive Steam Power Plant — External View.

mizer consists of a series of **pipes** placed in the path of the furnace gases, between the boiler setting and the stack. Draft is controlled by an **engine-driven fan** (not shown), placed between the economizer and the stack, to draw the gases from the furnace through the economizer and discharge them into the stack. The economizer raises the temperture of the feedwater to about 230 or 250 deg. fahr. By the addition of this heat-saving equipment the temperature of the feedwater is raised from 90 deg. fahr. to 250 deg. fahr., thus saving a large amount of heat which would otherwise be wasted. The thermal efficiency may thus be increased to 15 per cent and, under the best conditions, to nearly 20 per cent.

9. Locomotive Steam Power Plant. — An external view of the locomotive steam power plant is shown in Fig. 4, and a vertical section through the boiler in Fig. 5. The plant is of the noncondensing type, and the equipment is not essentially different from that installed in the same type of stationary plant. The space in which the equipment is installed is restricted, however, and the equipment must be accommodated to the space. The thermal efficiency of the best modern locomotive plants is from 5 to 8 per cent.

The boiler is a fire-tube boiler of the internally fired type; that is, the furnace is within the boiler

shell. An intensely hot fire is produced in the furnace, by a strong draft caused by exhausting the steam from the engine cylinders through a nozzle, which discharges the steam into a short stack attached to the smoke box. This reduces the pressure in the smoke box and causes an increased flow of air through the coal on the grate. Coal is stored in the U-shaped portion of the tender formed by the inner walls of the water storage tank, and is generally fired into the furnace by hand, except on large freight-hauling locomotives on which mechanical stokers are often used. The gases rising from the burning coal pass through the tubes to the smoke box.

Ashes from the **shaking grate** fall into an ash pan located directly under the grate. The amount of air admitted under the grate is controlled by dampers attached to each end of the ash pan and operated by rods and levers from the engineer's cab.

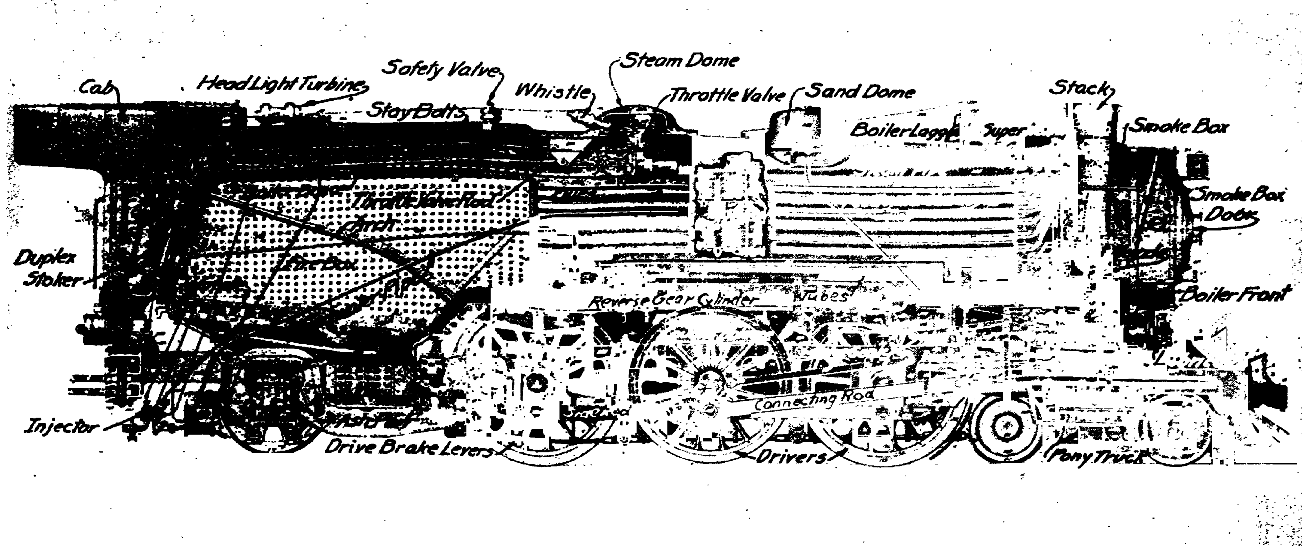
Steam generated in the boiler is led through a steam pipe that runs inside the boiler from the steam dome to the smoke box. The steam pipe here branches, a branch going to each cylinder of the engine. The flow of steam is regulated by a throttle valve operated by a lever in the engineer's cab. Feedwater, from the water-storage tank in the tender, is forced into the boiler by an **injector,** that is, a pump operated by a jet of steam.

Feedwater heaters are seldom used, because of lack of space. However, the necessity of saving fuel has caused them to be installed in the more recent locomotives. Superheaters, consisting of a system of piping in the smoke box and flues, are being used on up-to-date equipment.

A **vertical steam-driven air-compressor** is supported at the side of the locomotive boiler. This air compressor supplies air for operation of the brakes and sometimes for the valve mechanism on heavy locomotives. A small steam-turbine-driven generator, attached to the top of the locomotive boiler or to the side of the frame, furnishes electricity for the headlight.

Strictly speaking, the boiler does not have a setting. Its front end is attached to a saddle formed by the cylinder castings. At the rear end, brackets riveted to the side of the fire-box rest upon the side frame of the locomotive. The outside of the locomotive is covered with blocks of 85 per cent magnesia, an insulating material, to reduce the loss of heat, and a sheet-metal covering is placed over the magnesia blocks.

Locomotives are designated by two systems, which have reference to the arrangement of the wheels. One method gives an arbitrary name to the arrangement of the wheels, such as **Pacific, Prairie, Mogul, Mallet, Santa Fe.** The more common method is the **Whyte system,** which classifies locomotives by giving the number of wheels on the forward truck, or **pony truck,** the number of **driving wheels,** and finally the number of wheels on the rear, or **trailing truck.** Thus, a 4-4-0 locomotive has 4



Courtesy Railway and Locomotive Engineering

FIG. 5.—Locomotive Power Plant—Sectional View.

front wheels, 4 drivers, and 0 rear wheels. Table 1 gives a few typical wheel arrangements, with the Whyte classification and the corresponding arbitrary name.

TABLE 1.—WHYTE'S SYSTEM OF LOCOMOTIVE CLASSIFICATION.

Symbol	Wheel Arrangement	Type	Service
0-4-0	△ OO	*4-wheel Switcher*	*Yard*
0-8-0	△ OOOO	*8-wheel Switcher*	*Yard*
2-6-0	△o OOO	*Mogul*	*Freight*
2-6-2	△o OOOo	*Prairie*	*Freight and Passenger*
2-8-0	△o OOOO	*Consolidation*	*Freight*
2-8-2	△o OOOOo	*Mikado*	*Freight*
2-10-2	△o OOOOO	*Santa Fe*	*Freight*
4-6-2	△oo OOOo	*Pacific*	*Passenger*
4-8-2	△oo OOOOo	*Mountain*	*Freight*
0-8-8-0	△ OOOO OOOO	*Mallet articulated*	*Freight*
2-8-8-8-2	△o OOOO OOOO OOOOo	*Mallet articulated (Triplex)*	*Freight*

10. Marine Steam Power Plant. — Like the locomotive power plant, the marine power plant must occupy a small space. The steam marine plant is run "condensing"; it uses a surface condenser on ocean steamships in order to have pure water for use in the boiler, since sea water is not suitable for boiler feed, because of the salt it contains. Jet condensers are used on lake and river steamers. Figure 6 shows the location of the equipment on a typical lake steamship, together with transverse elevations through the engine and boiler rooms. The compact arrangement of the apparatus is evident.

The boilers are of the marine type and have fire-tubes or water-tubes. Those shown are of the fire-tube type. Coal is generally fired by hand, although stokers of the underfeed type are sometimes used. Oil is burned as a fuel under the boilers, in many ships.

The connection between the stack and the furnace is short. The stack is made of sheet steel; since it is relatively short as compared with a land installation, a fan, located as shown, is necessary to produce sufficient draft to burn the fuel. The fan may force air, under pressure, into the boiler room or ashpit of the boilers, or it may withdraw the gases by suction and deliver them to the stack. These methods are termed, respectively, **forced draft** and **induced draft.**

The engines are **vertical marine engines** and they drive the propellor shaft direct; that is, the engine shaft is directly connected to the propellor shaft, with a **thrust bearing,** placed at the engine end of the propellor shaft, to transfer the thrust of the propellor to the frame of the ship. When a turbine is used, the speed is reduced to that required by the pro-

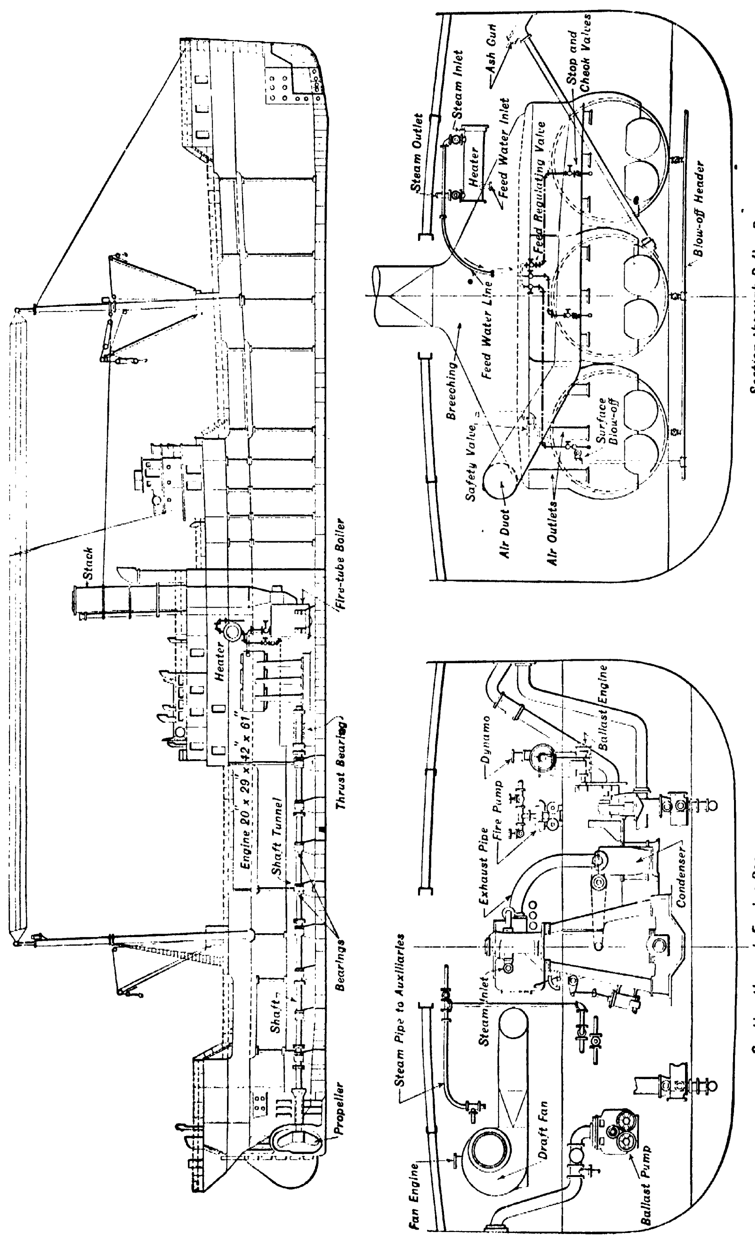


FIG. 6.—Marine Steam Power Plant.

pellor shaft; it is then necessary to use either **mechanical reduction gearing** or a **motor** connected to the propellor shaft and driven by current from the main turbo-generator.

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REVIEW QUESTIONS

1. Name the essential equipment in (*a*) a non-condensing power plant, (*b*) a condensing plant.

2. Distinguish between an isolated power plant and a central station plant.

3. Make an outline sketch showing the relative location of the essential equipment in a steam power plant operating (*a*) non-condensing, (*b*) condensing.

4. About what per cent of the heat in the coal is actually converted into power in (*a*) a condensing plant, (*b*) a non-condensing plant.

5. Name the heat-saving equipment which is often used in a steam power plant.

6. Trace the course of the feedwater from the hot well to the boiler, Fig. 2.

7. Trace the course of the steam from the boiler to the hot well, Fig. 2.

8. In what respect does the steam power plant of a locomotive differ from a stationary steam power plant?

9. What type of steam power plant is generally used on ships? Why?

CHAPTER II

STEAM BOILERS AND SETTINGS—BOILER AND SETTING FITTINGS

11. Foreword. — The general appearance of the modern boiler is not essentially different from that of a boiler of the same type manufactured fifty years ago. The principal improvements have been in the methods of construction.

There has, however, been great improvement in boiler operation, resulting from the use of furnaces of better design and construction. The present tendency in boiler operation is toward the use of higher steam pressure. A few years ago, a gage pressure of 100 pounds per square inch was considered high. To-day, many **power boilers** are operating at 275 pounds, and some boilers at 350 pounds, per square inch gage pressure in regular station practice. Manufacturers are prepared to furnish boilers capable of carrying 1200 pounds per square inch gage pressure and it seems probable that pressures as high as 1000 pounds per square inch may be used in important power stations before many years have passed.

Boilers carrying 25 pounds per square inch gage pressure, or less, are usually classed as **heating boilers.**

12. Material. — The material used in boiler construction should conform to requirements stated in the A. S. M. E. (AMERICAN SOCIETY OF MECHANICAL ENGINEERS) BOILER CODE.* Steel used in the boiler shell must be flange or firebox steel made by the open-hearth process. Its tensile strength must be from 55,000 to 65,000 pounds per square inch, and its crushing strength 95,000 pounds per square inch. When bent cold around a pin having a diameter equal to twice the plate thickness, the test piece should show no cracks. Firebox steel must be used when the part is to be under pressure and exposed to the fire or hot gases. *Cast iron should not be used for flanges, nozzles, fittings or valves, when under pressure, with a steam temperature exceeding 450 deg. fahr.*

13. Classification. — A classification of steam boilers is difficult, because of the large number of variations in a few fundamental types. The following classification is as satisfactory as any:

1. According to form	Plain cylindrical Flue Tubular Sectional

* The A. S. M. E. Boiler Code may be obtained by addressing the Secretary of the American Society of Mechanical Engineers, 29 West 39th St., New York City.

2. According to location of furnace	Externally fired Internally fired
3. According to use	Stationary Portable Locomotive Marine
4. According to direction of principal axis	Horizontal Inclined Vertical
5. According to relative positions of water and hot gases	Water-tube Fire-tube

14. Typical Steam Power Boilers. — In order to give a clear idea of boiler details, the following types and makes, which include all types of boilers generally used for power purposes, will be described:

1. Locomotive Boiler

2. Stationary boilers, as exemplified by the	Fire-tube	Horizontal return tubular Vertical tubular
	Water-tube	Babcock and Wilcox Heine Stirling Wickes
3. Marine boilers, as exemplified by the	Fire-tube —	Scotch Marine
	Water-tube	Babcock and Wilcox Emergency Fleet Corporation Dyson express

15. Boiler Nomenclature. — The following terms are common to the various types of boilers, and their application should be thoroughly understood. The application of each term is illustrated in Fig. 7.

Shell. — The boiler shell consists of several steel plates bent into cylindrical form and riveted together. Each separate cylindrical ring is called a course. A shell made up of two rings is called a two-course shell, and one made of three rings, a three-course shell. The ends of the shell are closed by means of boiler heads, which are made of flat, convex or concave boiler plate having flanged edges to permit riveting to the shell. The shell, together with the heads, forms the **drum.**

Setting. — The boiler setting is usually made of brick. It provides a means of support for some types of boilers and, at the same time, forms the walls of the furnace and combustion chamber.

Grate. — The grate consists of cast-iron bars or plates upon which the fuel is burned. The area of the surface upon which the fire rests is the

grate surface, and is usually expressed in square feet. A grate 5 feet wide and 6 feet long would have 30 square feet of grate surface.

Furnace. — The furnace, or firebox, is the space above the grate and below the boiler shell, and is enclosed by the side and front walls of the setting. It is the space in which the fuel is burned.

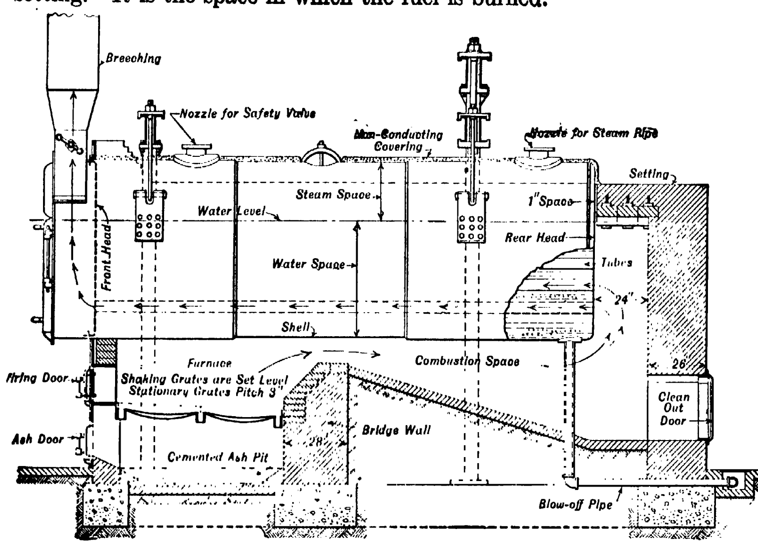


FIG. 7. — Boiler and Setting.

Combustion Space. — The combustion space is the space in which the volatile matter and combustible gases are burned.

Water Level. — The water level is the level at which water stands in the boiler shell. It is subject to considerable fluctuation, but for best operation the variation in level should be small. The height at which water stands in the boiler shell is shown by a **gage glass** located where it may be easily seen, or by **gage cocks** attached to the **water columns** or directly to the boiler head or shell.

Water and Steam Space. — The water space is the volume of the shell that is occupied by the water. The space occupied by water and tubes in a return tubular boiler is about two-thirds the volume of the entire shell. The steam space is the volume of the entire shell not occupied by water and tubes. It should have a volume of 0.65 to 1.00 cubic feet per boiler horsepower. (For definition of boiler horsepower, see Art. 198, page 187.)

Disengaging Surface. — The disengaging surface is the area of the water surface from which steam is separated. This area should be sufficient to prevent water from being carried from the boiler along with the steam. Steam carrying a large amount of water is said to be **primed.**

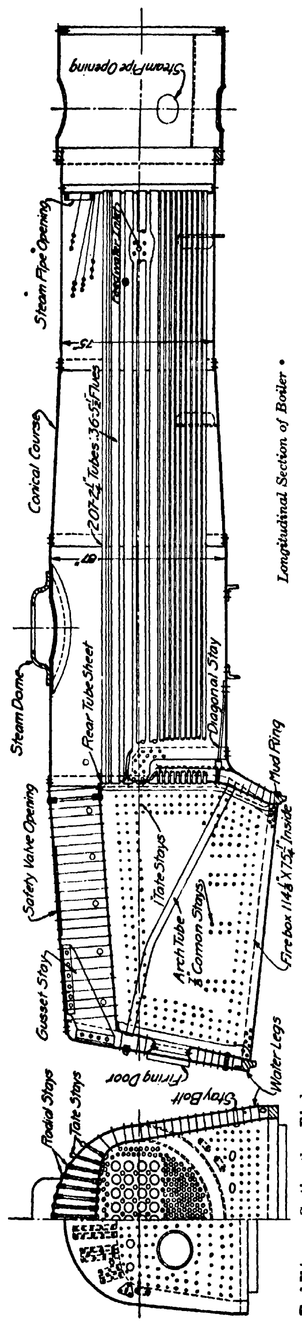


FIG. 8.—Conical Boiler with Wide Firebox for Pacific (4-6-2) Locomotive.

Heating Surface. — The **water heating surface** is that surface of the boiler which is exposed to hot gases on one side and water on the other. **Superheating surface** is that surface having steam on one side and hot gases on the other. Heating surface is expressed in square feet and is generally calculated for the mean water level.

16. Locomotive Boiler.* — A longitudinal and a transverse section of a modern locomotive boiler are shown in Fig. 8. The locomotive boiler is a *straight fire-tube boiler having an internal firebox.* It requires a large amount of heating surface and a grate upon which coal can be burned at a rapid rate. This is obtained by using a large number of staggered tubes and a strong draft induced by the steam exhausted from the engine cylinders. Locomotive boilers are commonly classified by the shape of the shell; as, **straight top,** having the cylindrical shell of uniform diameter from the firebox to the smoke box; **wagon top,** having the steam dome over the firebox and a sloping course from the firebox to the cylindrical shell; **extended wagon top,** having one or more cylindrical courses between the firebox and the sloping course which tapers on the top and sides to the diameter of the main shell; **conical,** Fig. 8, hav-

* The various parts which form a fire-tube boiler are described in Art. 16 in order to make the discussion as complete as possible, and only those parts which differ are considered under other types of boilers.

ing one or more cylindrical courses between the firebox and the sloping course and a conical connecting course which tapers to the diameter of the main shell.

The shell, or **barrel,** shown in Fig. 8, consists of three courses, which are fastened to each other by lap joints, No. 1, Fig. 9. The seam of rivets thus made around the circumference of the boiler is known as a **ring,**

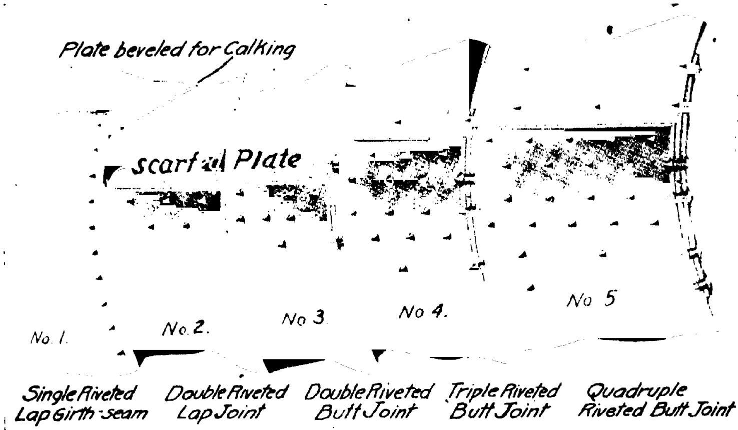


FIG. 9.—Types of Boiler Joints.

or **girth seam.** The free ends of the plate forming each course are brought together to form a **butt joint,** as shown in No. 3, Fig. 9. A **covering strip,** or **welt,** is placed outside, and a wider covering strip, or welt, inside. These covering strips hold the ends of the plate together by means of rivets. The joints uniting the ends of each course are called **longitudinal joints** and must be arranged as described. A butt joint may be either double, triple, or quadruple riveted. Only ring seams are lap riveted.

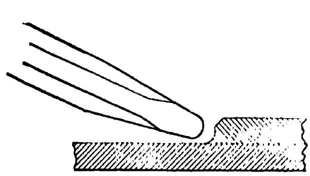


FIG. 10.—Round Nose Calking Tool.

The joint made by fastening No. 1 and No. 2, Fig. 9, together illustrates how the outside strap is **scarfed,** or **drawn out,** and carried under the ring seam to produce a tight joint where the longitudinal and ring seams meet. The ends of all outside plates are beveled at an angle of 15 degrees before they are rolled into shape. After being riveted, the edges are **calked;** that is, hammered against the plate to which they are riveted, thus making a joint that will not leak. For calking, a tool, Fig. 10, having a rounded edge is used. Sharp-edged tools should not be used, as they may injure the plate.

The **firebox** is riveted to the rear course of the shell. It derives its name from the fact that it forms the chamber within which the fuel is burned. It also forms a support for the grates, which are at the bottom of the firebox. *The firebox consists of inside, front, rear and side sheets riveted together and to the* **crown sheet,** which forms the top of the firebox. The crown sheet and side sheets are made of one piece, whenever possible. The outside sheets are separated from the inside sheets by a space, forming the **water leg,** the lower end of which is closed by a steel ring called the **mud** or **foundation ring.** The front outside sheet is riveted to the rear course of the shell and is known as the **throat sheet.** The outside top or **roof sheet** and the side sheets are generally made in one piece. The outside rear sheet is flanged inward and is riveted to the roof and side sheet. There is a flanged opening in this sheet for the fire door. **Handhole openings** on a level with the crown sheet and **clean-out plugs** above the mud rings are provided for cleaning purposes. *The best heating surface of the boiler is that which surrounds the firebox.*

Locomotive boilers are often classified according to the construction of the firebox, as **wide firebox, narrow firebox, Wootten, Belpaire,** and **Jacobs-Shupert.**

The wide firebox illustrated in Fig. 8 rests on the frames and extends out beyond the driving wheels at the sides, while the narrow firebox extends down between the frames and the driving wheels. The Wootten firebox is very wide and shallow and has a curved crown sheet of large radius. It is used on locomotives, burning anthracite coal, to give a large grate area.

The Belpaire firebox, Fig. 11, has a flat crown sheet joining the side sheets by curves of short radii. The roof sheet and upper part of the outer side sheets are flat and parallel to those of the inner firebox.

The Jacobs-Shupert firebox is one in which the usual arrangement of flat sheets is replaced

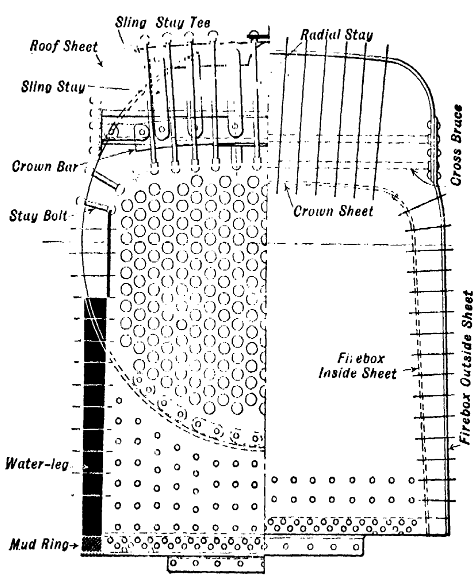


FIG. 11. — Belpaire Firebox.

by inner and outer sets of channel-shaped sections riveted together, with flanges away from the fire. Stay bolts are replaced by **stay sheets,** one at each joint of the channels and secured by the same rivets that hold the channels. The stay sheets are partially cut away in the water leg to allow horizontal circulation of the water. All seams and joints are submerged.

The front inside sheet of the firebox forms the **rear tube sheet;** it contains openings for the tubes by which it is connected to the **front tube**

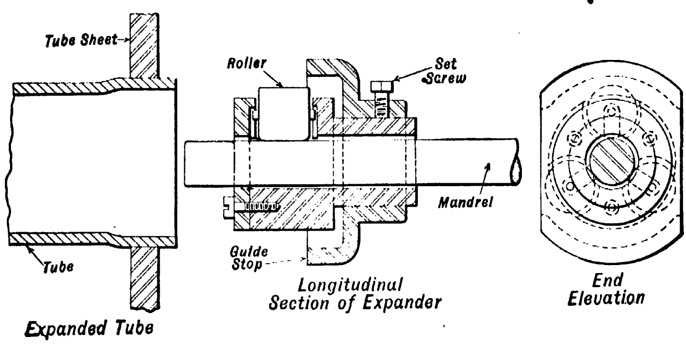


Fig. 12. — Dudgeon Expander.

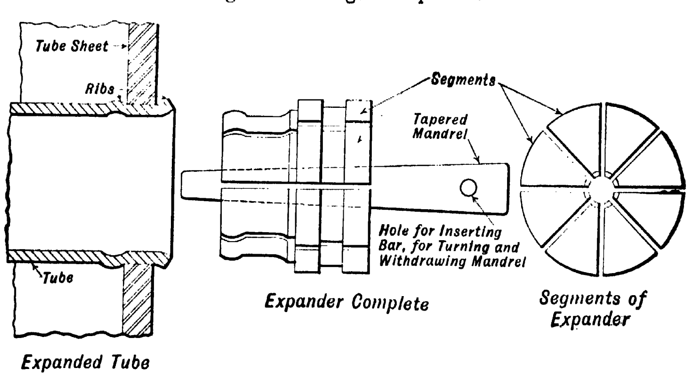


Fig. 13. — Prosser Expander.

sheet which is flanged outward and riveted to the front course. The latter also contains an opening for the steam pipe passing to the cylinders. The **tubes** are made of the best grade of charcoal iron and have an outside diameter of 2 to $3\frac{1}{2}$ inches.

Tubes as large as $5\frac{1}{2}$ inches are sometimes used, and are then known as **flues.** *Locomotive boilers have from 300 to 500 tubes and flues,* which extend about $\frac{3}{8}$ inch through openings in each head before beading. The holes in the tube sheet for the tubes are made $\frac{1}{8}$ inch larger than the outside

diameter of the tubes, and the tubes are expanded to a tight fit in the tube sheets by some form of expander. Figure 12 shows the **Dudgeon** and Figure 13 the **Prosser** expander, together with the appearance of the tubes after being expanded by each type of expander. After expansion, the projecting ends of the tubes are rounded over against the plate by a **beading tool,** Fig. 14. **Stay tubes,** which are heavier than ordinary tubes, are sometimes used. They may be threaded at the ends and often have the firebox end protected by **ferrules.**

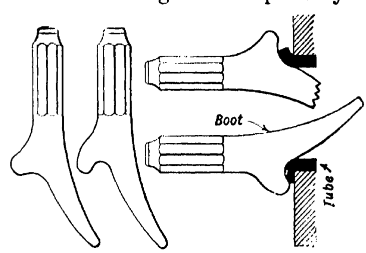


Fig. 14. — Beading Tool and Method of Use.

To the front course of the shell, an additional ring, known as the **smoke box,** is riveted, the front of which is closed by a casting called the **smoke-box front.** The smoke pipe or stack is attached to the top of the smoke box, and the openings for steam pipes are at the bottom. There are screens in the smoke box which prevent the escape of large cinders. On top of the shell and in front of the firebox is an opening over which is a dome-shaped chamber known as the **steam dome.** It is made of sections of curved steel plates riveted to the shell. Steam is taken from this elevated part of the boiler in order that it may contain as small an amount of water as possible. A **sand dome,** having no opening into the boiler and holding sand for use under the driving wheels, is attached to the shell in front of the steam dome.

Stays or braces are used to prevent failure of flat surfaces when under pressure. **Radial stays,** Fig. 8, are round steel bars with both ends threaded and may be used to brace the crown sheet. They are usually set radial to the curvature of the crown and roof sheets and are screwed into the surfaces to be stayed. Nuts may be placed on the projecting ends, or the ends may be riveted over.

The front end of the crown sheet is often supported by a **sling stay,** Fig. 11, which permits slight relative movement of the supported and supporting surfaces and prevents breaking of stays. It consists of straps of steel fastened at each end to an **angle** or **tee bracket,** by a pin joint. A bracket is riveted to each surface to be stayed, and short pieces of pipe are placed around the rivets between the lower bracket, or **crown bar,** and the crown sheet to allow water to circulate between the bracket and plate.

Stay bolts, Fig. 15, are used to brace the flat surfaces forming the water legs. The stay bolt ordinarily used on locomotive boilers is like the radial stay except that it is shorter. A hole $\frac{3}{16}$ inch in diameter is often

drilled in the outer end of each stay bolt to a depth slightly beyond the inside of the plate. A broken stay bolt may then be detected by the escape of steam and water through the opening. Flexible stay bolts may be used to allow relative movements of the two plates which they secure,

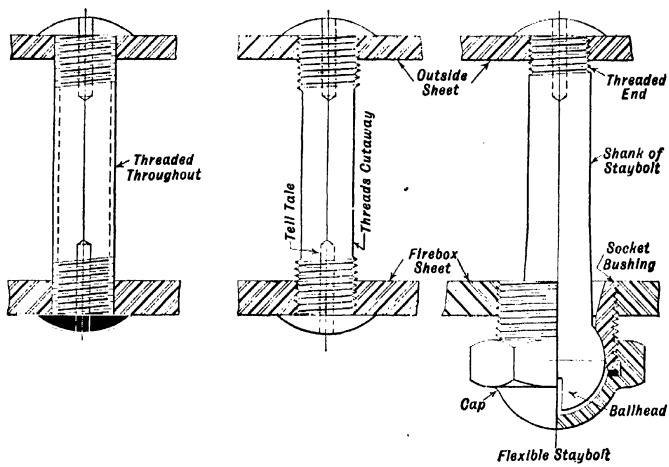


FIG. 15. — Common and Flexible Stay Bolts.

and thus reduce the number of broken stay bolts caused by unequal expansion of the inner and outer plates of the firebox.

Diagonal stays, Fig. 16, brace that part of the front head above and below the tubes, and the outside rear firebox sheet above the crown sheet. These stays are made of steel and are riveted to the shell and head. **Gusset stays,** Fig. 16, are often used in place of diagonal stays. The gusset stay is a steel plate riveted to angle irons, which are riveted to the stayed surfaces.

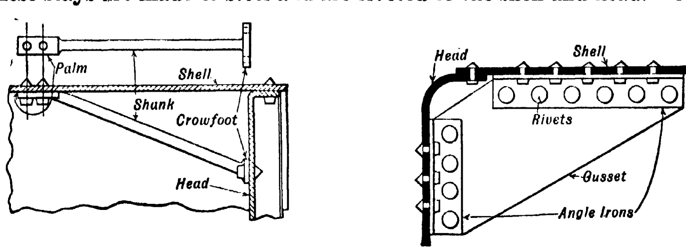


FIG. 16. — Diagonal and Gusset Stays.

A safety valve, which prevents the pressure in the boiler from becoming dangerously high, is placed in the top of the steam dome, or is fastened to a nozzle riveted to the top of the outside firebox sheet. A **fusible**

plug, another safety device, is screwed into the crown sheet and protects the boiler against low water level. A **blow-off pipe** is attached to the outside of the water leg at its lowest point.

17. Horizontal Return-tubular Boiler. — This type of boiler is commonly used in small power plants. As its name implies, the gases

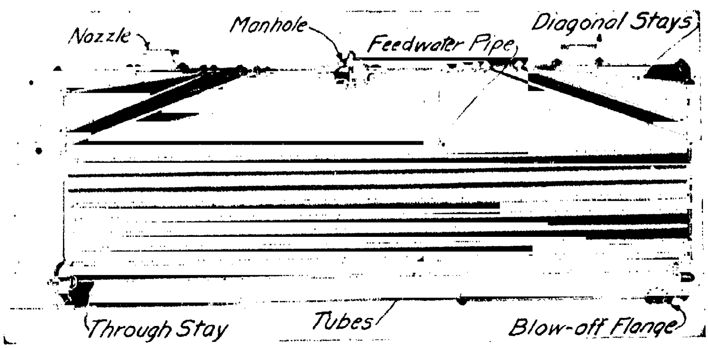


FIG. 17. — Longitudinal-section H. R. T. Boiler.

from the furnace, after passing below the boiler along the outside of the shell to the rear, return through tubes to the smoke connection at the front of the shell. A longitudinal-section of this boiler is shown in Fig. 17, and an external view in Fig. 18.

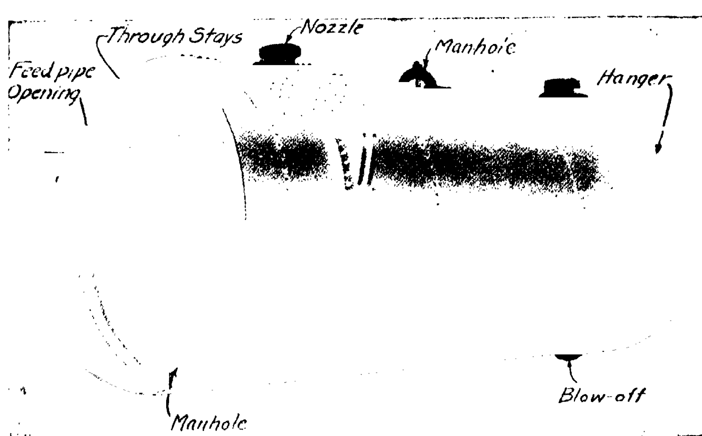


FIG. 18. — Outside View of H. R. T. Boiler.

The shell is made of three courses with a smoke connection attached to that part of the front course which remains after the front top portion is cut away to make an opening for the passage of the gases to the breeching.

The flange of the front head is turned outward, to permit riveting by machine and make construction easier. On the so-called New York boiler, this flange is turned in. The rivets forming the joint are thus protected by water from possible overheating, but riveting is made more difficult during construction. A **manhole** or **handhole** is located in the front head, below the tubes, to give access to the lower part of the shell for inspection or cleaning, and the tubes are arranged in the head as shown in Fig. 18, to give as free a passage for water circulation as possible.

The area of the heads above the tubes may be stayed by (1) diagonal stays, or (2) heavy angles or channels and through braces, Fig. 40, page 48. The angles or channels support the boiler head with rivets, and are in turn supported by through braces from head to head. These braces, or longitudinal stays, pass through the boiler heads and are threaded for nuts inside and outside of the heads. That part of the head area occupied by the tubes does not require bracing, because the tubes act as braces. The area of the heads below the tubes is braced by a special form of through brace, as shown in Fig. 17. It is attached to the rear head by means of a pin connection through two angles, which are riveted to the head, but are separated from it by nipples in order to prevent overheating of the plates.

The use of the diagonal brace below the tubes is not permitted, because the pads or enlarged flattened shoulder of the brace, when riveted to a shell, which is surrounded by hot gases, would not conduct heat away with sufficient rapidity to prevent the shell from being overheated.

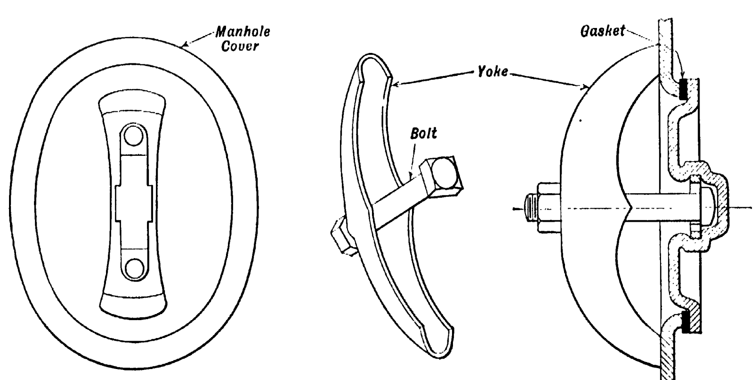


Fig. 19. — Manhole and Cover with Yoke.

Steam nozzles, made of cast iron or pressed steel, are riveted to openings in the top of the front and rear courses. A safety valve is attached to one nozzle, and the boiler stop valve and main steam line to the other. An 11 by 15-inch elliptical manhole, which gives access to that part of

the boiler above the tubes, is located in the middle course at the top of the shell. A **manhole ring** is riveted to the shell, or the boiler plate is flanged over, as shown in Fig. 19, and machined to form a seat for the manhole cover, which is held in position by bolts and yokes. A **gasket,** made of lead or rubber, is generally placed between the manhole ring and the flat edge of the manhole cover to make a tight joint.

The **boiler feed pipe** is screwed into a **brass** or **steel bushing,** Fig. 20, at one side of the front head, or into a **flanged connection** on top of the front course. An internal brass feed-pipe is screwed into the bushing or flange and is supported by a strap from the boiler shell or from a through brace. This pipe passes toward the rear, turns across to the middle of the shell and discharges downward with an open end between the central row

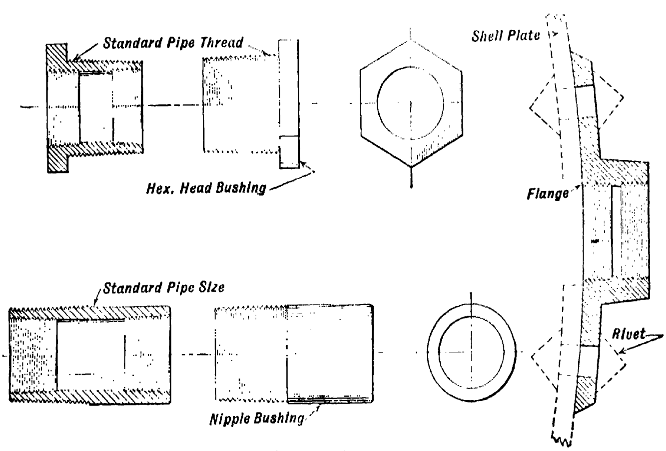


FIG. 20. — Typical Feedwater Pipe Bushings.

of tubes below the water level, and about three-fifths of the length of the tubes from the front head.

The height of the water in the boiler is shown by a water column at the front of the setting, as shown in Fig. 21. The lowest water level should not be less than $3\frac{1}{2}$ inches above the top row of tubes.

Scale and sediment, which collect in the bottom of the shell, are blown out through a blow-off connection at the bottom of the rear course of the shell. The **blow-off flange** is riveted to the shell and the blow-off pipe screwed into the flange.

The boiler shown in Fig. 21 is supported by brackets riveted to the upper half of the boiler shell. The brackets must be arranged in pairs on each side of the boiler, on each end course, near the ring seams of the middle course. The front brackets rest on a plate in the setting. The front end

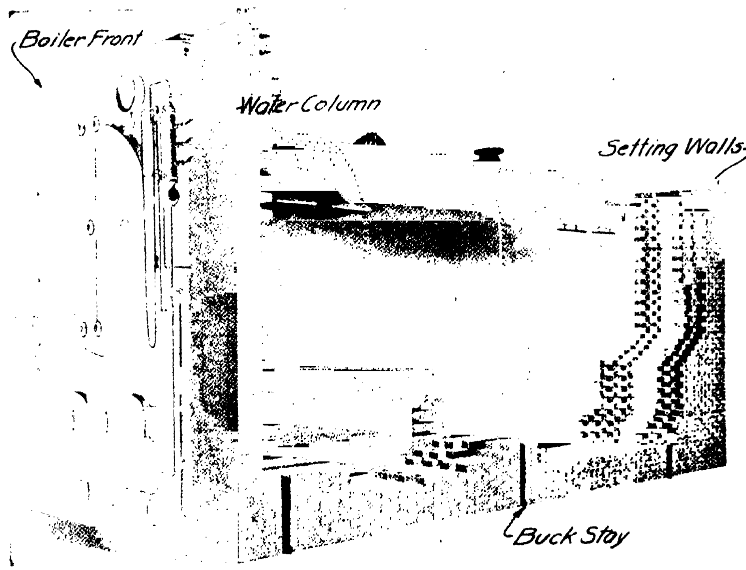


FIG. 21. — H. R. T. Boiler supported by Brackets.

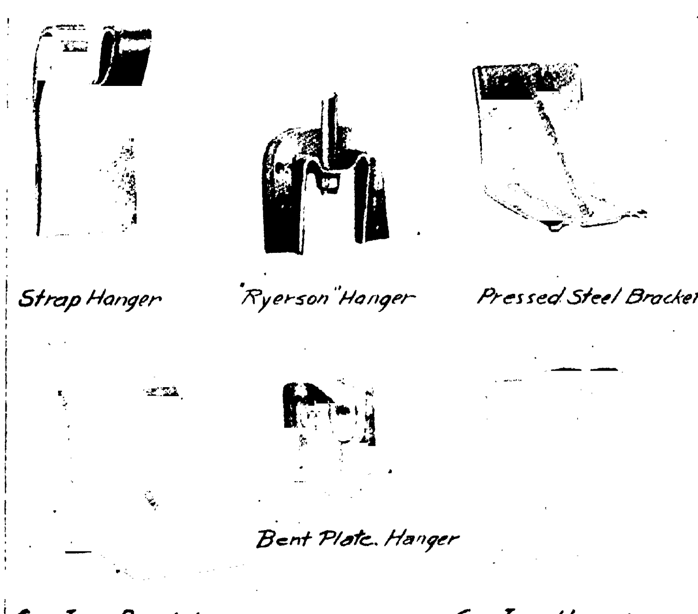


FIG. 22. — Typical Hangers and Brackets used to support H. R. T. Boilers.

is fixed, and when the boiler expands, the rear end moves on rolls placed between the bracket and a plate resting on the setting.

Types of hangers and brackets used to support horizontal tubular boilers are shown in Fig. 22. The pressed steel bracket or hanger is the most satisfactory.

Steel columns and I-beams built into the setting, Fig. 23, are also used to support this type of boiler. The boiler is hung from I-beams by rods which hook into hangers riveted to the upper part of the boiler shell.

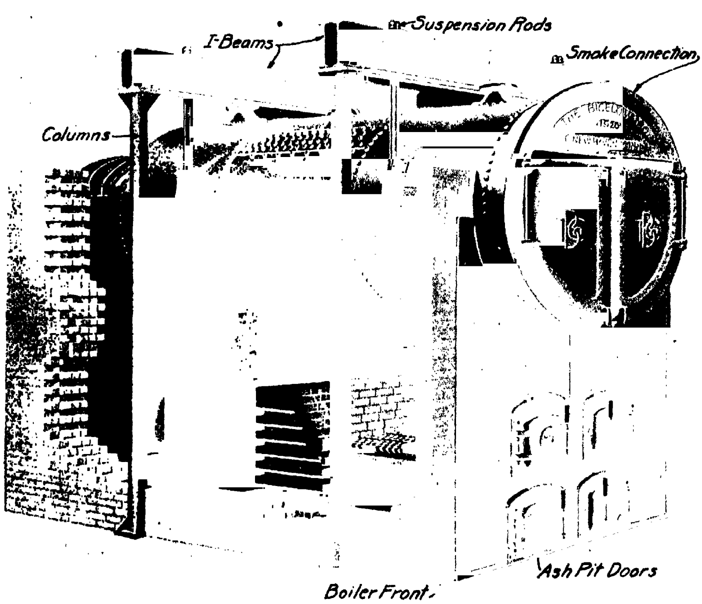


FIG. 23. — H. R. T. Boiler supported by Hangers and Columns.

The second method is preferable to the first, as the boiler is entirely free from the setting and will not injure it when contracting or expanding. A third method of support often used is known as the three-point suspension. This method, proposed by Mr. Woolson, is most satisfactory, as the boiler is evenly supported. The front of the boiler is suspended by hangers and hanger rods on each side of the shell. The suspension rods at each side of the rear course of the boiler are swung from the ends of an equalizing lever, which is supported at its center by the I-beam overhead construction.

18. Setting for Horizontal Return-tubular Boiler. — Boiler settings are ordinarily made of brick. Such settings are comparatively porous and often crack, thus permitting air to leak into the furnace. Settings are also made of **silocel,** an insulating brick of low heat conductivity. Silocel is made from a rock which contains silica and, as constructed, has a high percentage of small air spaces. The brick may also be covered with a steel casing lined with magnesia block riveted to the plate, or by a covering of wire and asbestos. The steel covering is expensive, but allows small air leakage, while settings made of silocel are liable to crack in much the same way as brick settings.

Top, front, and side views of a brick setting are shown in Fig. 24. The setting rests upon a concrete foundation which should have sufficient depth

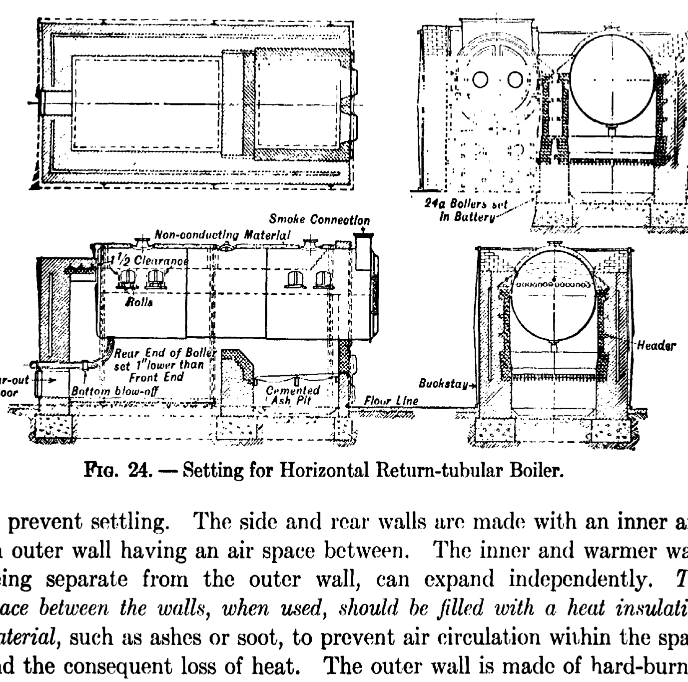


Fig. 24. — Setting for Horizontal Return-tubular Boiler.

to prevent settling. The side and rear walls are made with an inner and an outer wall having an air space between. The inner and warmer wall, being separate from the outer wall, can expand independently. *The space between the walls, when used, should be filled with a heat insulating material,* such as ashes or soot, to prevent air circulation within the space and the consequent loss of heat. The outer wall is made of hard-burned strong brick and the inner wall, which comes in contact with the hot gases, is made of **firebrick,** a material especially prepared to resist high temperature. Firebrick wall is often laid with every sixth row of brick as a header, to permit repair of the wall between any two headers without disturbing the remainder. The walls may be made solid and their total thickness varies from 16 to 24 inches, depending upon the size of the boiler. The side walls are carried above the center line of the boiler as shown in Fig.

24, and the space between the boiler and inner wall is filled with asbestos material to prevent leakage from or into the furnace. The top of the boiler is covered with non-heat-conducting material. The return-tubular boiler is ordinarily set 1 inch lower at the rear than at the front, with *sufficient clear space left in front of the setting to permit removing and replacing defective tubes.* When two boilers are placed in a setting having a common wall separating the furnaces, or in battery, the construction is as shown in Fig. 24*a*.

The furnace shown in Fig. 24 is externally fired; that is, the fire is external to the boiler shell. The grates are supported by **bearer bars** fastened to the side walls and are set from 3 to 5 inches lower in the rear than at the front. The distance between the boiler shell and the grate should be at least 30 inches for anthracite coal and 36 inches, or preferably more, for coal which gives off large quantities of gas.

At the rear of the grate a **bridge wall** extends between the side walls, and projects a distance above the level of the grates. This wall

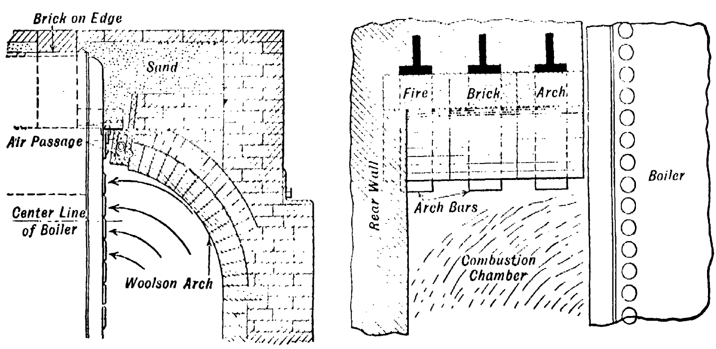


FIG. 25. — Two Methods of making Back Connection to Boiler.

often supports the rear end of the grate, retains the fuel on the grate, and assists in mixing the gases and air passing over it. The connection between the bridge and side walls should be so made that there will be room for expansion of the bridge wall; otherwise, cracked side walls will result. The top of the bridge wall should be about 12 inches from the shell for anthracite coal, and the distance should be increased for coals that evolve large amounts of gas.

The distance between the rear wall and the rear of the shell varies from 18 to 24 inches. The rear wall is carried above the top row of tubes, and the space between the rear wall and shell is bridged over, to prevent escape of the furnace gases, by a horizontal or curved firebrick wall, or arch, strengthened and supported by T-bars. As the expansion of the boiler may be as much as $\frac{1}{2}$ inch, a movable connection is made between the shell

and the wall. Two methods of making this connection are shown in *Fig. 25, the first of which is known as the* **Woolson arch.**

The front wall consists of two **fire-door arches** made of hard-burned fireclay or of tile with a cast-iron fire-door arch. The **arch** rests upon a dead plate made to hold firebrick, or upon a bearer plate supported by the side walls. Above the fire-door arches is a hard brick wall having a firebrick lining. Every other layer of firebrick, between the fire-door arch and the shell, is arranged as a header.

A **clean-out door** in the rear wall gives access to the inside of the setting, for cleaning. The blow-off pipe is carried through a **thimble** in the rear wall, and the space in the thimble not occupied by the blow-off pipe is filled with asbestos wool, to prevent leakage from or into the combustion space.

19. Setting Fixtures. — Certain castings and forgings, Fig. 26, are needed to complete the setting. A **cast-iron** or **steel casing** covers the

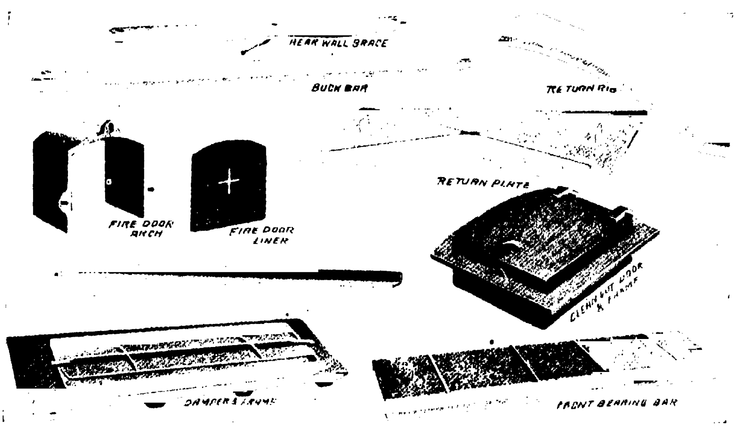


Fig. 26. — Setting Fixtures.

front of the boiler and setting. This casing may be a one-half, three-quarter, or full-front casing. In an **overhung front setting,** the smoke box extends out from the boiler front over the fire doors, as in Fig. 23, page 29. In this respect the overhung front setting differs from the **flush front setting.**

The **rear-wall brace** and **buckstays** are used to prevent spreading of the rear and side walls. The **front** and **back bearing bars** support the grate **bars, with** the front bearing bar also serving as a support for the fire-door **arch.** The **return plate** and **rib** are used in making the connection between the boiler and the rear setting wall. The **fire-door liner** protects the fire door from the direct heat of the fire and thus prevents

it from overheating and warping. The **damper** and **damper frame** are *located in the breeching* connection, and the damper turns upon trunnions carried by the breeching or the damper frame.

20. Vertical Tubular Boiler. — This type of boiler requires small floor space per boiler horsepower and is well adapted to carry the steam pressures now used. It furnishes superheated steam, that is, steam at a temperature above that corresponding to the steam pressure. It is built in sizes from 50 to 500 boiler horsepower.

A vertical tubular boiler of the Manning type is shown in Fig. 27. The shell is made of three courses, the lower course serving as the outside sheet of the firebox. A smoke box is attached to the top course, and has an opening to which the breeching connection is attached. Between the lower outside course and the middle course, a reduction in size is made by means of a **reverse flange,** or **Ogee ring.** This permits a smaller diameter of shell with no decrease in the area of the firebox. *The inside sheet of the firebox is fastened to the outside sheet by stay bolts, which are the only stays used in the Manning boiler.* The reverse flange is a source of weakness and, to do away with it, the shell is often made straight from top to bottom, or a tapered course is placed between the outer sheet of the

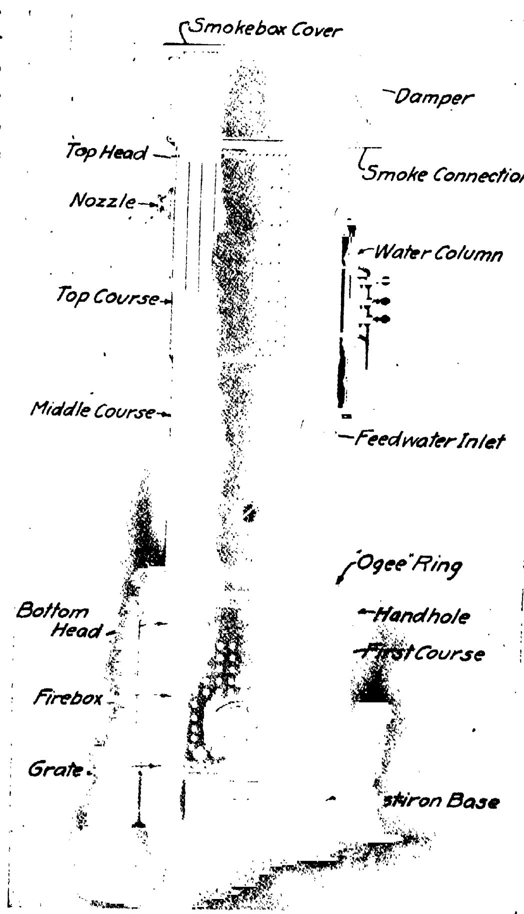


FIG. 27. — Manning Vertical Fire-tube Boiler.

firebox and the next course. These types of construction are known as **straight-shell** *and* **tapered-shell vertical tubular boilers.** The lower part of the water leg is closed by a wrought-iron mud ring, while handholes are provided just above the mud ring for cleaning purposes.

The lower head forms the crown sheet, and is the support for the lower ends of the tubes. It is riveted to the top of the firebox sheet. The upper head is riveted to the upper course of the shell, with the flange turned outward. The tubes vary from 2 to $2\frac{1}{2}$ inches in diameter. They are expanded into each head and are well beaded to give maximum support to the crown sheet.

Handholes are provided on a level with the crown sheet, opposite each row of tubes, for removing dirt and sediment which may have collected on the crown sheet. The fusible plug is screwed into an extra thick tube on an outside row, about one-third the length of a tube above the crown sheet; and the opening in the shell for removing and replacing the plug is directly opposite. The outer sheet is re-inforced by a steel band where the handholes are located. Tapered and straight-shell types generally have manholes in the middle course. A fire-door opening is flanged in the outer firebox sheet, and the inner sheet is riveted to it in such a manner that a leaky connection at this point is avoided.

The water column is attached to the front of the boiler, and the steam nozzle to the upper part of the top course of the shell. The external feedwater pipe is screwed into a flange riveted to the upper part of the middle course, and an internal distributing pipe, running between a row of tubes, is screwed to the flange from the inside. A blow-off connection is attached to the lower part of the water leg.

This type of boiler does not require the usual setting. A cast-iron base, resting on a concrete foundation, forms the ashpit arch and serves as a support for the boiler and also for the circular grate. A brick base is often used instead of a cast-iron base. In this case the brick base is made about 20 inches high, and a plate, which supports the grate and upon which the boiler rests, is placed upon the brick.

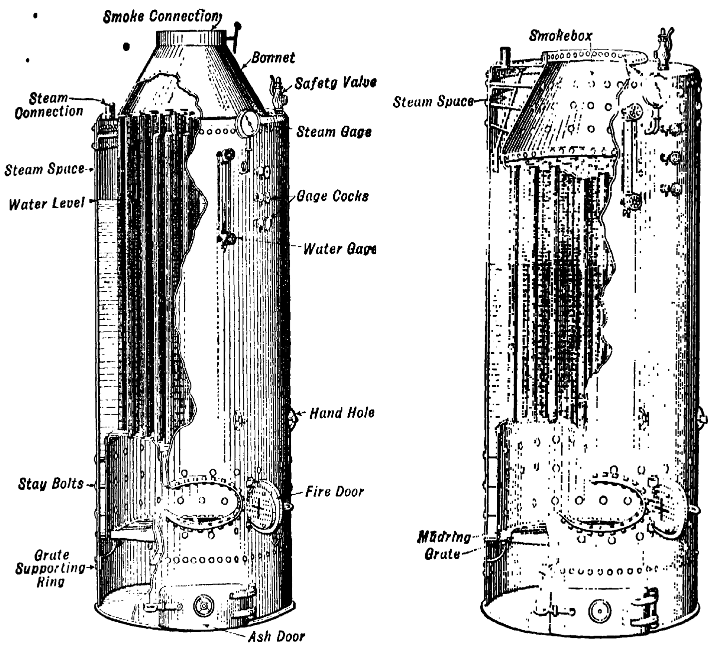
The furnace of the vertical tubular boiler is internally fired, the hot gases coming into contact with the tubes and the inside plates of the firebox. The outside plates may then be made sufficiently thick to withstand high pressures, and the inside firebox plates, which are made thinner to transmit heat readily, are easily stayed from the outer shell.

The gases pass straight up through the tubes to the smoke box. The water-heating surface consists of the inside area of the box and as much of the tube surface as is surrounded by water. The tube surface above the water level is superheating surface.

21. Small Vertical Tubular Boiler. — This type of boiler is made in sizes from 3 to 100 horsepower and is used on construction jobs where

a portable boiler is necessary. The steam space is small because the water level comes within a short distance of the top head.

Two types of small, vertical fire-tube boilers made with a one-course shell are shown in Fig. 28. The firebox in each type is formed as in the larger types of vertical fire-tube boilers. The regular upright type fre-



(a) Full Length Tubes (b) Submerged Tubes

FIG. 28. — Small Vertical Fire-tube Boilers.

quently gives trouble from tubes leaking at their point of connection to the top head. This difficulty is overcome by using the submerged type, in which the tubes are surrounded by water throughout their length. This design gives a restricted area at the top, for separation of steam from the water; and the cone forming the smoke chamber above the tube sheet, being subjected to pressure, sometimes becomes leaky. A water column is not ordinarily used on this type of boiler, the water level being determined by gage cocks and a water glass attached directly to the shell.

22. Water-tube Boilers. — Water-tube boilers have water inside the tubes and hot gases surrounding the tubes. They may be classified as follows:

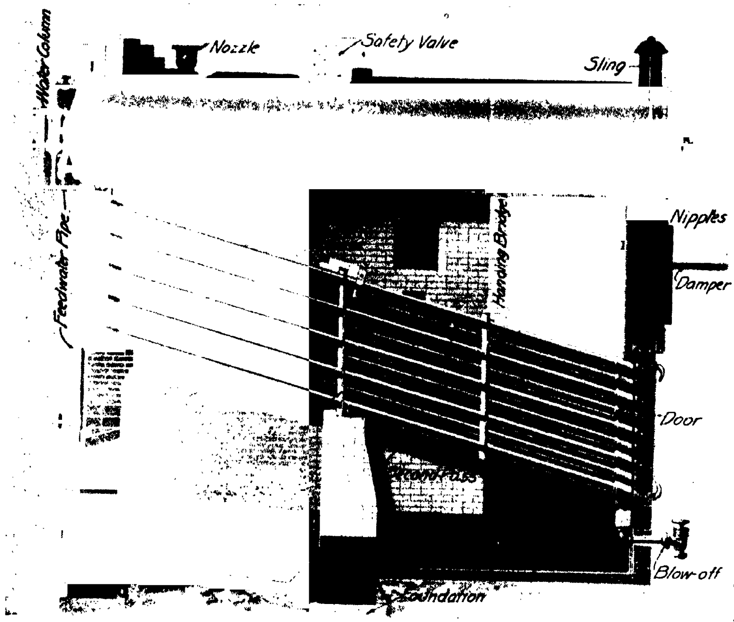


FIG. 29a. — Babcock and Wilcox Longitudinal-drum Water-tube Boiler.

FIG. 29b. — Front View of B. and W. Boiler showing Headers.

1. According to service	Stationary Marine
2. According to position of drum	Vertical Cross Longitudinal
3. According to form of header	A header for each row of tubes A common header for all tubes
4. According to arrangement of tubes and drums	A series of inclined bent tubes connecting the drums

23. Babcock and Wilcox Longitudinal-drum Water-tube Boiler. — This type of boiler, Figs. 29*a* and 29*b*, is made with one or more horizontal drums having a cross box, Fig. 30, riveted to the end courses of each drum and connected to a series of headers by short tubes, or nipples, expanded into openings in the headers and cross boxes. A single row of tubes connect each front and rear header, and the capacity of the boiler depends upon the number of headers and tubes used. In recent large-capacity units, the drums extend crosswise with respect to the tubes, and the boiler is called a **cross-drum water-tube boiler.** Each drum is generally made of three courses constructed as described in the article on the

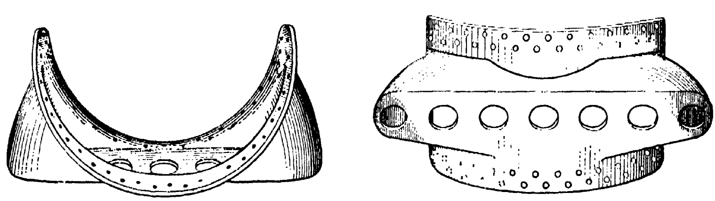


FIG. 30. — Forged Steel Cross Box.

locomotive boiler. The heads are forged by hydraulic press and are dished to a radius equal to the diameter of the shell. When so dished the heads are seldom stayed, although it is considered better practice, by some engineers, to use stays.

A manhole is flanged in at the center of each head, and the flanged edges are machined to a flat surface. The manhole plates are machined to fit the manhole opening and are held in place by forged steel yokes and bolts. Flat surfaces are provided in the front head for water column and feedwater connections.

Two steam nozzles are generally used. They are riveted to the shell and located as shown. The tubes are expanded into headers having a serpentine form, Fig. 31, and the ends are flared but not beaded. This shape of header arranges the tubes so that they are staggered. The headers may be made of cast iron or steel, the choice of material depending upon the pressure which the boiler is to carry, and may be vertical, as in

Fig. 29*a*, or inclined. Opposite the end of each tube an elliptical or circular handhole is located. It must be of sufficient size to permit cleaning or removal of a tube, and, if the handhole covers are to be removed through the circular handholes, one handhole must be made larger than the others.

The handhole covers are forged plates, fitted inside, and shouldered to center in the opening. A nut and yoke, or a special clamp and nut,

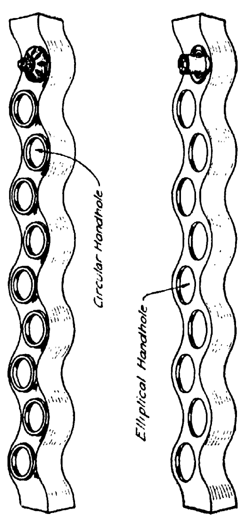
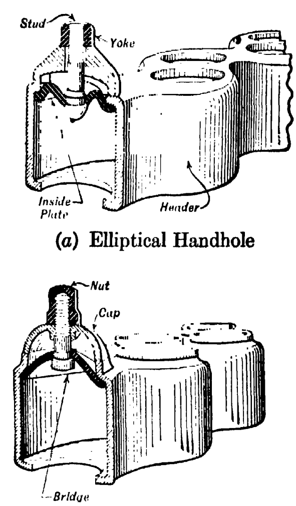


FIG. 31. — Vertical Steel Headers.



(*a*) Elliptical Handhole

(*b*) Circular Handhole

FIG. 32. — Types of Handholes for Wrought Steel Header.

Fig. 32, hold the covers in place. The joint between the hole and cover is made tight either by having the surfaces milled true and the joint made metal to metal or by placing a thin gasket between the surfaces.

A forged steel **mud drum** $7\frac{1}{4}$ inches square is attached to the bottom of the rear headers by means of wrought-iron nipples expanded into the mud drum and the headers. The mud drum is tapped for blow-off connections, and handholes are provided for cleaning.

The boiler is suspended from a steel-girder frame by steel rods passing around each end of the drum and is thus entirely independent of the brickwork of the setting. This method of support permits free expansion and contraction without injury to the boiler or brickwork, and also makes it possible to repair the brickwork without disturbing the boiler or its pipe connections.

The feedwater pipe enters the front head as shown in Fig. 33 and opens into the boiler several feet from the front head. The water circulation

is from front to rear of drum, downward through connecting tubes to the rear headers, then forward through the tubes to the front headers, and up into the drum again. A **deflecting baffle** is placed above the header connections at the front, to throw back the water from the steam, since the steam formed in the passage through the tubes is liberated when it reaches the front of the drum.

The steam formed in the boiler passes to the steam pipe line through a **dry-pipe,** which is a perforated pipe attached to the steam nozzle inside the boiler. Its purpose is to prevent the water carried by the steam from passing into the pipe line.

24. Setting for Babcock and Wilcox Boiler. — The setting for this type of boiler resembles the setting described in Art. 18, page 30. It is higher, because the horizontal drum has to be hung at a height suitable for the grade of fuel to be burned. Ordinarily, the distance from the grate to the lowest row of tubes should be at least 6 feet. The side and front walls of the setting, together with the bridge wall, enclose the furnace space. The bridge wall can be located to give any depth of furnace demanded by the fuel to be burned.

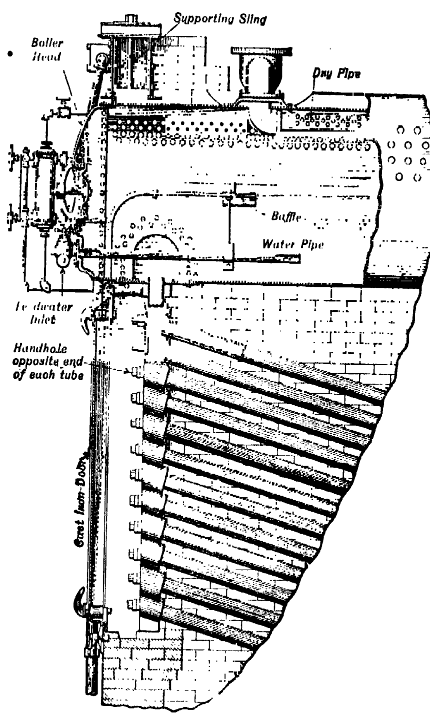


FIG. 33. — Section showing Construction of Front End of B. and W. Boiler.

Baffle plates or **walls** separate the combustion chamber into compartments called **passes.** The front baffle wall, Fig. 29*a*, is supported by the tubes at the rear wall of the furnace, and the rear baffle wall is attached to a hanging bridge wall, which is supported by a special **cross girder.**

The front baffle forces the gases to pass around the forward portion of the tubes to a chamber beneath the drum or drums. The gases then turn downward over the front baffle into the central portion of the tubes, called the **second pass,** until they pass around the lower end of the sec-

ond baffle into the **third pass** and across the rear portion of the tubes to the damper box and flue connection in the rear wall.

The baffle walls are formed of cast-iron baffle plates lined with special firebrick and held in position by clamps. A baffle wall made of firebrick tile, with a plastic filling, Fig. 34, may be used.

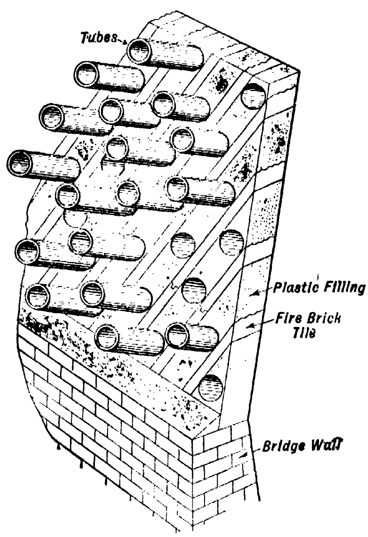


FIG. 34. — Turner Baffle Wall.

The space between the bridge wall and the rear boiler wall, and beneath the tubes, forms a pocket into which much of the soot from the gases in their passage through the second pass is deposited.

Leakage of air into the furnace through the space between the header sections is prevented by filling this space with asbestos cement or asbestos rope.

Cleaning doors, Fig. 29*a*, page 36, are provided, to give access to the tubes for cleaning purposes. Small **dusting doors** are located in the side of the setting, to permit the cleaning of all parts of the heating surface, and sufficient space is allowed between settings for this purpose. The front of the boiler is enclosed by an ornamental cast-iron front, Fig. 29*b*. This figure also shows the top supporting girder and the method of making the water connection when two boilers are set in battery.

25. The Heine Water-tube Boiler. — The Heine water-tube boiler, Figs. 35 and 37, is typical of a large number of makes. It consists essentially of *one or more drums connected to water legs or headers with tubes connecting the headers.*

The construction of the shell is the same as that already described for the Babcock and Wilcox boiler. The shells are made in diameters from 30 to 48 inches and in length from 17 to $21\frac{1}{2}$ feet. The main steam nozzle is attached to the front course of the shell, and a throat opening for the water-leg connection is cut in the bottom of the shell near each end. **Throat stays** are riveted across this opening to compensate for the metal thus cut away. The feedwater pipe passes through the front head and empties into a **sheet-steel mud drum,** running parallel to the shell and near its bottom. This mud drum is closed, with the exception of a small opening at the top, near the front end, and impurities which

may be in the water are deposited in the mud drum and are removed through a blow-off pipe passing from the mud drum through the rear head.

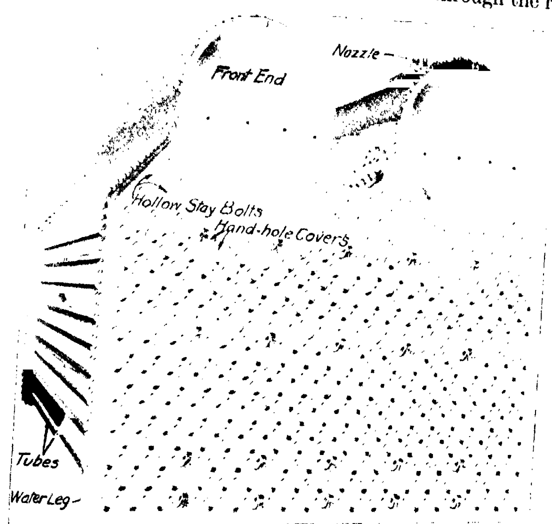


FIG. 35. — Heine Two-drum, Two-pass, Water-tube Boiler from Front Water Leg.

Over the throat opening at the front of the shell is a deflection plate closely fitted to the head and the sides of the drum. This plate serves to throw down the water which is carried by the steam when the steam is liberated from the water. The dry pipe is directly above this plate.

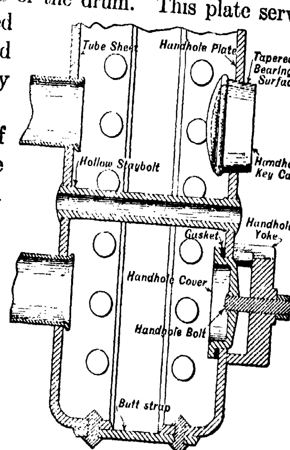


FIG. 36. — Water Leg of Murray Boiler.

The water legs, Fig. 36, are made of two flat plates, called respectively the tube sheet and the handhole sheet, which are flanged and joined together, except at the top, by a "butt strap." *The flat surfaces of the water legs are stayed by hollow steel staybolts*, screwed into each sheet and riveted over against the sheet. The tubes are fastened to the tube sheet by being expanded and then slightly flared to increase their holding power. The bottom of the rear water leg is connected to the blow-off piping.

A recent type of **handhole cap,** used on this boiler to replace the ordinary type of handhole cover, is also shown in Fig. 36. The claims for this type of cap are that it does not require a gasket, is self-tightening and non-leaking, and saves time and labor when the boiler is cleaned. The water column is attached at the top to the upper part of the front head and at the bottom to the top of the front water leg.

26. Setting for Heine Boiler. — As ordinarily set, the tubes and drum incline downward from the front to the rear with a pitch of ½ in 12, Fig. 37. *The front end is supported on heavy cast-iron columns,* which serve

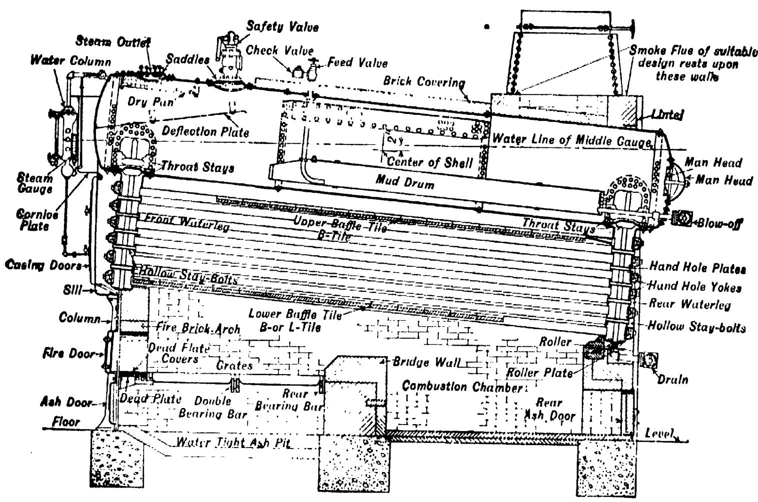


FIG. 37. — Section of Heine Boiler and Setting with Names of Parts.

as a support for the fire and ash door frames, and other castings which make up the boiler front. When the furnace is stoker-fired this end is hung from girders and the columns are omitted. The front wall of the setting is lined with firebrick and is directly back of the front castings. *At the rear the water leg rests on rollers,* which in turn bear on iron plates set in the top of the low rear wall of the setting, thus allowing for expansion. The space between the rear header and the wall is filled with asbestos fiber to prevent leakage of air into the furnace. The side walls are of solid brick lined with firebrick and carried up to the height of the ornamental front. **Returns** are made which follow the curvature of the water legs and shell, being carried by properly anchored rods secured to rolled steel buck-staves.

On the lower row of tubes, extending back within 3 or 4 feet of the rear end, is placed a firebrick baffle tiling, and similar tiling is placed on

the upper row of tubes extending lengthwise from the rear to within 3 or 4 feet of the front end. The gases of combustion pass lengthwise of the tubes between the baffles and thence along the under side of the shell to the smoke connection at the rear.

The direction of circulation of the water is the same as in the Babcock and Wilcox boiler.

Access to the setting, for cleaning, is obtained through a rear cleaning door and two side cleaning doors at the top of the setting. Soot and dust is blown from the tubes by a **soot blower** consisting of a number of small steam or air nozzles inserted into the hollow stay bolts. The stay bolt-holes not occupied by the soot-blower nozzles are closed by wooden or cast-iron plugs.

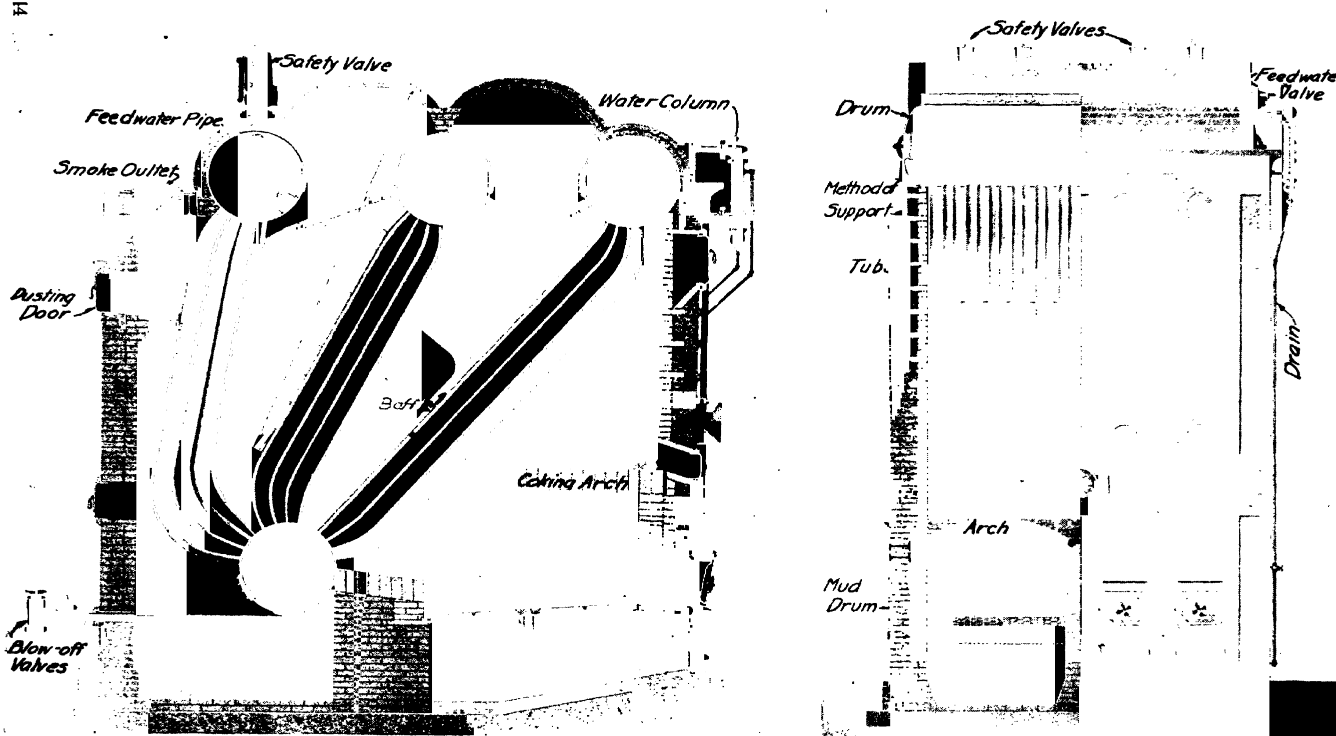
27. Stirling Water-tube Boiler. — This boiler is typical of many makes, such as the Badenhausen and Erie City, *which have one or more parallel steam drums above connected by sloping or vertical tubes to one or more parallel water drums below.* This type of boiler has a large amount of heating surface, obtained by a large number of tubes, and consequently is suitable for high capacities. The drums are generally made of a single course with the ends closed by dished heads containing the manholes. The tubes are bent to enter the drums radially and are expanded to fit the tube holes, but are not beaded.

A section, taken through the four drums and the walls of the setting of a Stirling boiler, is shown in Fig. 38. The three upper drums contain water and steam and are set at the same level. They are connected to the mud drum by tubes, so curved as to enter the tube sheets radially. The center drum is equidistant from the front and rear drums and its steam space is connected to the steam spaces of the front and rear drums by a row of curved **steam-circulating tubes.** The water spaces of the front and center drums are connected by rows of water-circulating tubes. The water space of the center drum is connected with the mud drum by one-half the tubes of the front row of the rear bank, which support a baffle protecting the rear steam drum.

The main steam outlet is placed on top of the rear drum, which also carries the safety valves. The feedwater pipe enters the rear of this drum and discharges into a **perforated trough,** which distributes the water over a relatively large portion of the area of the drum.

The water column is attached to one end of the center drum and the blow-off pipe to the bottom of the mud drum.

Each drum is made of a tube sheet, riveted by butt-and-strap longitudinal seams to a drum sheet. The drum heads are of forged steel, one head in each drum being provided with a manhole. The upper drums are supported at both ends by lugs resting on a rectangular structure of rolled steel and entirely independent of the brickwork; the lower drum is sus-



Side Elevation showing Arrangement of Tubes and Drums. Partial Front View and Section through Front Drum.

FIG. 38. — Stirling Water-tube Boiler.

pended from all of the steam-and-water drums by the water tubes, swinging entirely free of the setting. The leakage of air around the ends of this drum is prevented by soft asbestos packing placed between it and the brickwork.

28. Setting for Stirling Boiler. — The entire boiler is surrounded by four walls having clean-out doors at the side of each bank of tubes, with the front of the setting covered by an ornamental cast-iron and steel front, as shown by the partial front elevation in Fig. 38. The brick walls of the combustion chamber are faced with firebrick, and a **coking arch** is sprung over the front of the grate, with the grate directly under the arch. The furnace gases are directed, by firebrick baffle tiles, to pass from the grate along the first bank of tubes, then down the middle bank and up the rear bank, passing out through the smoke connection at the rear.

The water, which is fed into the rear steam-and-water drum, passes downward through the rear bank of tubes to the mud drum, thence upward through the front bank of tubes to the front steam-and-water drum. The steam formed during the passage upward through the front bank of tubes becomes separated from the water in the front drum and passes through the steam circulating tubes into the middle drum and then, with the steam generated in the middle bank of tubes, into the rear drum, from which it passes through the dry pipe into the steam main. The water from the front drum passes through the water circulating tubes into the middle drum and thence downward through the middle bank of tubes to the mud drum, from which it again passes up the front bank to retrace its course.

29. Wickes Vertical Water-tube Boiler. — A Wickes boiler is shown in Fig. 39. It consists of *an upper and a lower drum joined by straight tubes about 22 feet in length.* The tubes are arranged in parallel rows, are expanded into the drum heads and are flared to increase their holding power. The top drum is the steam drum. It contains a manhole in the top drum head, a steam nozzle, and feed pipe and water column connections. The bottom drum is the mud drum. The boiler is supported by brackets riveted to the mud drum and resting upon plates fastened in the walls of the setting. The blow-off connection and a manhole are located in the lower head of the mud drum. This boiler does not require stays.

30. Setting for Wickes Boiler. — The entire boiler is enclosed by a circular brick setting having a firebrick lining. A firebrick baffle wall extending nearly to the upper drum divides the setting vertically into a front and rear compartment. The tubes in front of this wall are called "**risers**" and those in the rear "**downcomers.**" The wall of the setting is prevented from spreading by bands of steel. The brickwork is often covered by sheet iron to make the setting air-tight. Doors are provided for cleaning pur-

poses and the fire and ashpit doors are supported by the ornamental front casing.

The gases from the furnace pass up the front side of the baffle wall to the

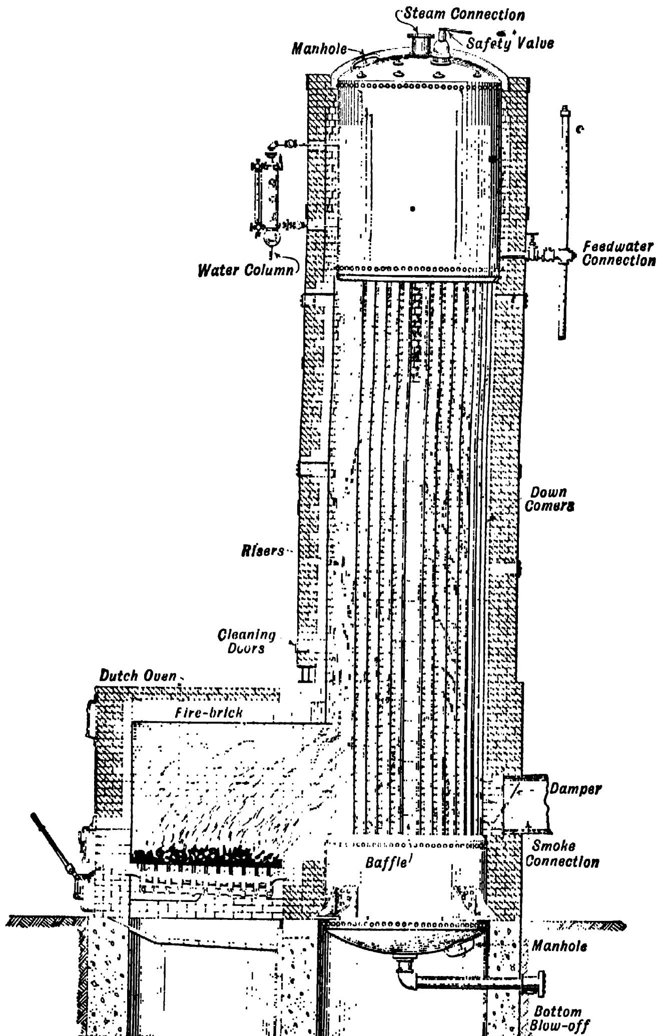


FIG. 39. — Wickes Vertical Water-tube Boiler.

bottom of the upper drum, then turn and pass down the rear side, through the smoke connection at the bottom of the setting, to the chimney.

The water circulation is from the bottom drum up the "risers," across the top drum, then through the "downcomers" to the mud drum. A deflection plate is placed in the top drum over the front set of tubes. The water level is about at the center of the upper drum and hence is always above the top of the tubes.

31. Marine Boilers. — Boilers used on shipboard are of two general types, fire-tube and water-tube. The water-tube type is used where rapid steaming and high pressure qualities are essential, as in naval vessels and rapid passenger service. The fire-tube boiler is used where extreme lightness or high speed are not essential, as in the merchant marine. The fire-tube boiler is used to the greater extent at the present time.

The most common type of marine fire-tube boiler is the Scotch marine boiler. It is self-contained, requires low head-room and is manufactured in units up to 2000 boiler horsepower.

There are many classes of marine water-tube boilers. The majority have one or more top cylindrical drums and one or more drums below. The upper and lower drums are connected by straight or curved tubes. The Almy, Yarrow, Thornycroft, and Dyson express boiler are typical makes. The Babcock and Wilcox marine boiler has the lower drum replaced by headers with connecting tubes. This make is known as the free-circulation type and is much used. Units as large as 4500 boiler horsepower are manufactured.

32. Scotch Marine Fire-tube Boiler. — Longitudinal and vertical half sections are shown in Fig. 40. The boiler consists of a two-course shell ordinarily varying in diameter from 7 to 16 feet and in length from 8 to 11 feet. The heads are made of two plates, an upper and a lower, riveted together and flanged for riveting to the shell. Heads up to 15½ feet in diameter are now being made from single plates.

The shell ordinarily contains from one to four cylindrical, corrugated-steel furnaces, 42 to 48 inches in diameter, attached at the front end to a flanged opening in the front head and at the rear end to the front wall of the combustion chamber, into which the furnace opens. These furnaces are internally fired, are entirely surrounded by water, and, because of the corrugations, require no staying. The grates and bridge wall occupy the front part of the furnace, and the space in each furnace beneath the grates forms the ashpit. At the rear end of the furnace is the combustion chamber, it is entirely surrounded by water and does not require a firebrick lining. The rear wall of the combustion chamber is flat and is supported from the rear head by stay bolts. The side walls of the combustion chamber conform to the curvature of the shell and are attached to it by stay bolts. The front wall forms the rear tube sheet. A crown sheet is riveted to the upper ends of the walls and forms the top of the combustion space. **Girder**

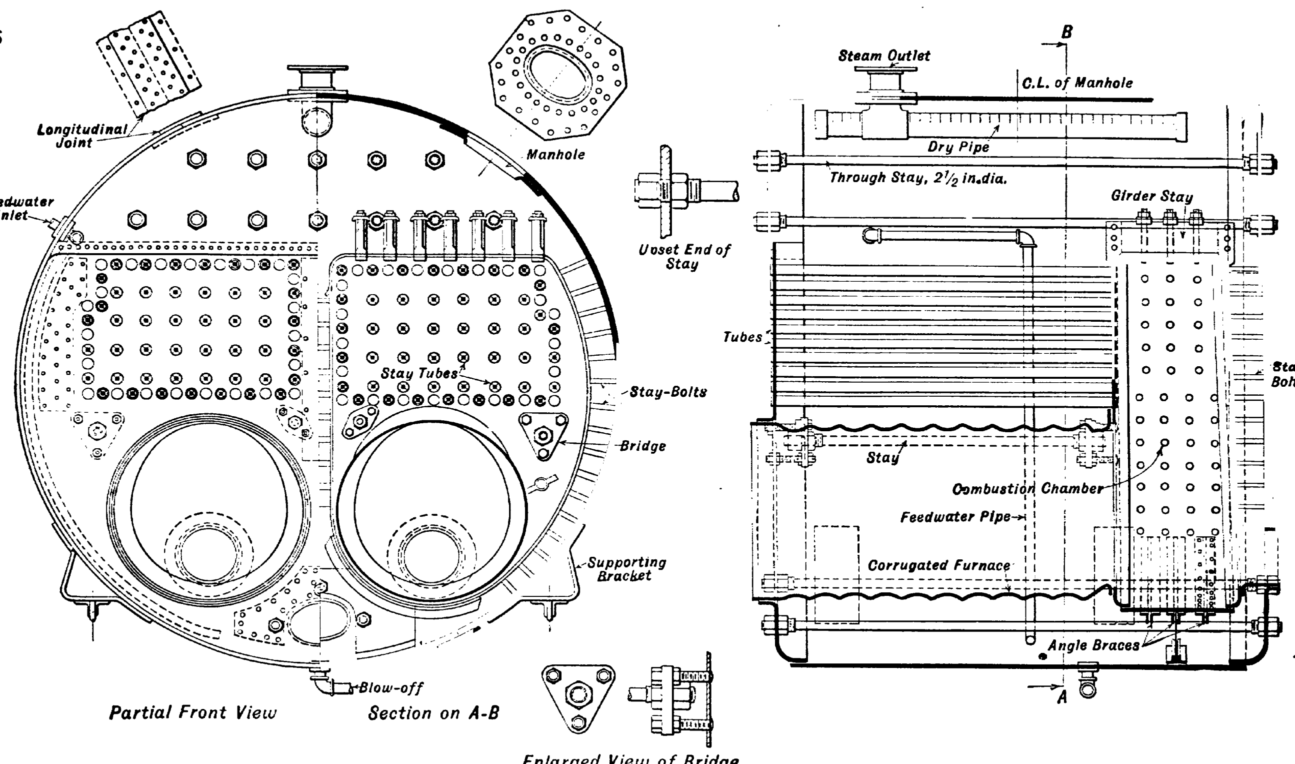


FIG. 40. — Scotch Marine Fire-tube Boiler.

stays or **crown bars** support the crown sheet. The walls of the combustion space are the best heating surface of the boiler.

The tubes which are arranged as shown in the half-sectional view, are expanded into the tube sheet and beaded. The area of the front and rear heads above the tubes is braced by channels and through braces. The area of the front head below the furnace is braced by a special form of through stay fastened at the rear end to the front wall of the combustion space by angle irons and a pin. Such bracing permits some flexibility.

The openings in the upper half of the boiler shell are for a manhole and a steam nozzle. The feed pipe enters the side of the shell and passes down

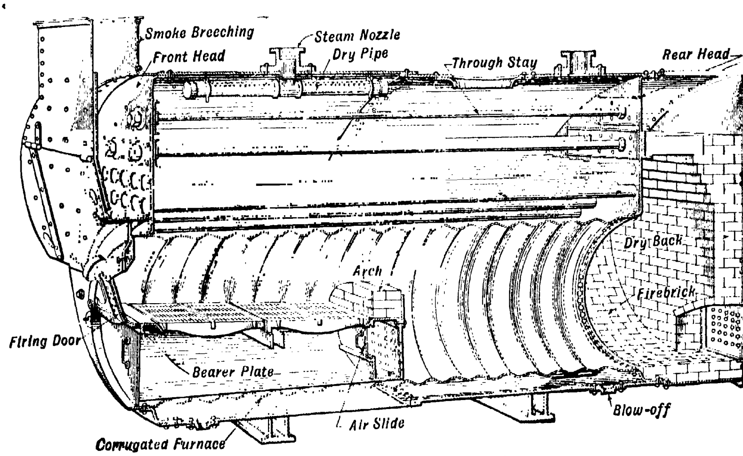


FIG. 41. — Dry-back Scotch Boiler.

around the side of the shell, discharging about midway the length of the boiler. The water column is generally placed in an inclined position, though the connections are the same as for any fire-tube boiler. Several manholes are provided in the front head, and there is an opening into the combustion chamber in the rear head.

This boiler *does not require a setting*. It is supported by a cradle, or **saddle,** securely fastened to the frame of the ship. Adjustable **turn-buckle** stays hold the boiler in place in the saddles.

The gases pass from the furnace over the bridge wall and into the combustion space, returning through the tubes to the smoke connection at the front.

The water level is about eight inches above the top row of tubes. The water circulation is down the sides and up the middle when two furnaces are in the shell. It is often necessary to force the circulation by a steam jet and a series of nozzles placed near the bottom of the boiler.

When more than one furnace is used, each may have a separate combustion space, or all may be connected to one combustion space. The boilers may be either single ended or double ended. A **double-end boiler** consists essentially of two single-end boilers placed back to back, with the back heads removed, the shells joined, and the rear sheets of the combustion chambers stayed together.

A modification of the Scotch marine boiler much used for office and hotel buildings, because of the low headroom required, is shown in Fig. 41. The combustion chamber is made of firebrick carried by an extension of the shell. This type is known as a **dry-back marine boiler.**

33. Babcock and Wilcox Marine Water-tube Boiler. — This boiler, Fig. 42, is constructed along the same general lines as the Babcock and Wilcox longitudinal water-tube boiler. The main difference is in its shape and in the arrangement of the heating surfaces. The drum is located at the front of the boiler above the headers and crosswise of the tubes. It is made of a shell having a single sheet with the heads dished and containing the manhole openings. The steam nozzle is located at the top of the drum with the main and auxiliary feedwater connections and gage glasses attached to the front side of the drum. The tubes are extra heavy and are arranged in sections, each section having a front and rear header. The front headers are connected to the bottom of the drum by short tubes four inches in diameter. The rear headers are set higher than the front headers so that the tubes, which are expanded and flared, have an upward slope of 15 degrees from front to rear. The top of each rear header is connected to the upper part of the drum by a 4-inch return tube, and the drum is reinforced at the points where the tubes enter by a strip of steel riveted to the inside of the shell. The side headers are carried down to a level with the grate, and the lower tubes are replaced by forged steel boxes 6 inches square,

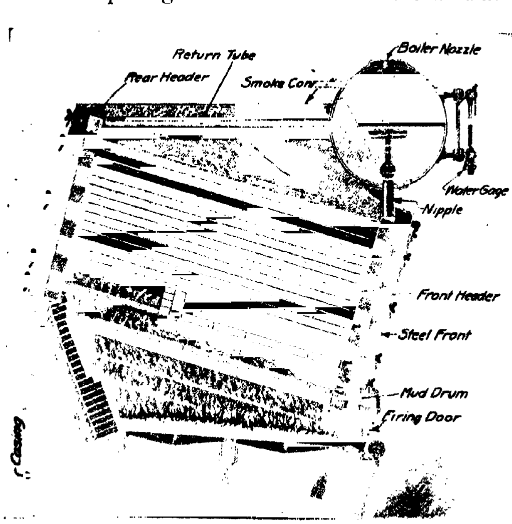


FIG. 42. — Babcock and Wilcox Marine Water-tube Boiler.

which form the side walls of the furnace and maintain a cool side casing. At the bottom, the headers are attached to a 6-inch square mud drum to which the blow-off connection is attached.

This boiler is supported by a structural iron framework, to which the outer casing is attached, and a brick setting is not required. The spaces between the outer row of tubes are filled with asbestos fiber and the whole covered with a steel casing to make an air-tight covering.

A firebrick wall extending from a level with the grate to the bottom of the rear headers makes the rear wall of the furnace. The gases from the furnace are forced to pass to the rear by a horizontal baffle of firebrick, along the lower row of tubes, then upward to the return tubes. Here a baffle plate deflects the gases downward between a set of baffle walls to the lower row of tubes. They then pass upward around the lower end of the front baffle wall to the drum and the breeching connection at the top.

The water circulation is down the front headers, or **down takes,** through the tubes to the rear headers, or **uptakes,** returning to the drum through the

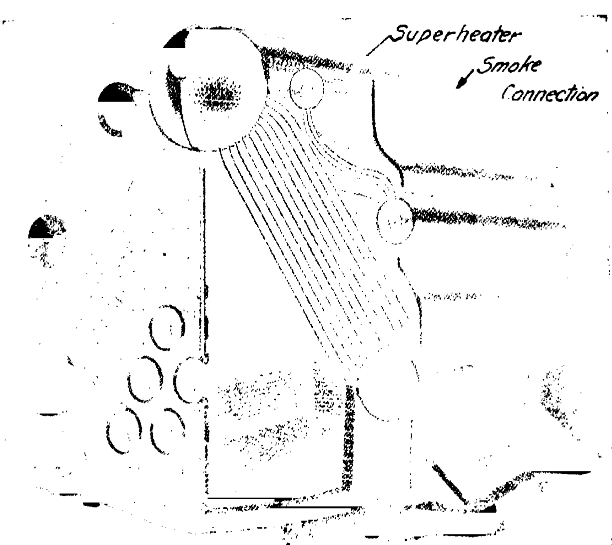


FIG. 43. — Dyson Express Type Water-tube Marine Boiler.

return tubes. A baffle plate at the open ends of the return tubes deflects the water and liberates the steam, which passes around the ends of the baffle plate and through the dry pipe to the steam nozzle. The water level stands at about the center of the drum.

Hinged to the framing at the front and rear of the boiler are large doors

giving access to the handhole plates. Dusting doors, with each opening covered by a shutter sliding vertically, are provided in the side walls, for use of a steam lance.

34. Emergency Fleet Corporation Marine Water-tube Boiler. — This boiler is of the cross-drum type and differs from the Babcock and Wilcox boiler mainly in the type of headers and baffling used. The header is of the type used on the Heine water-tube boiler. (Art. 25, page 40.) There are numerous boilers of this general construction.

35. Dyson Express Type Marine Boiler. — This type of boiler is illustrated in Fig. 43. It has a single drum at the top with two drums at the bottom. The upper drum is connected to each of the bottom drums by a large number of small curved tubes. The furnace is located underneath the tubes and, as shown, is arranged to burn oil. The small drums are superheater drums. The gases pass among the tubes to the smoke connection at the top.

36. Comparison of Fire- and Water-tube Boilers. — The advantages and disadvantages of these fundamental types of boilers may be summed up as follows:

Fire-tube

Advantages	Disadvantages
1. Small floor space.	1. Small steam space, pressure liable to fluctuate.
2. Low first cost.	2. Not easily cleaned or examined.
3. Ruptured tubes easily replaced.	3. Liable to leak at ends of tubes at stays and corners of firebox.
4. Large water storage capacity.	4. Pressure reduction necessary in time.
	5. Size limited to 200 hp.

Water-tube

Advantages	Disadvantages
1. Steam can be raised quickly.	1. More sensitive to change in conditions.
2. Suitable for high pressures.	2. More difficult to deal with leaky tubes.
3. Generally safer.	3. Must have good feedwater.
4. Less weight for large capacity.	4. Liable to prime.
5. Can be readily forced.	

37. Boiler Fittings. — A new boiler is provided with certain attachments, often called fittings, accessories, or trimmings.

The fittings commonly supplied are a **water column, fusible plug, safety valve, steam gage, and blow-off and feedwater connections.**

38. Water Column. — An external and sectional view of a water column is shown in Fig. 44. It is essentially a hollow cast-iron vessel having two connections to the boiler. The top connection enters the steam space of the boiler, either through the front head or through the top of the front course of the shell. The bottom connection enters the water space at

least 6 inches below the lowest permissible water level. The connecting pipes should be at least 1 inch in diameter, with the water connection made of brass. Valves are not ordinarily allowed in the water-column connec-

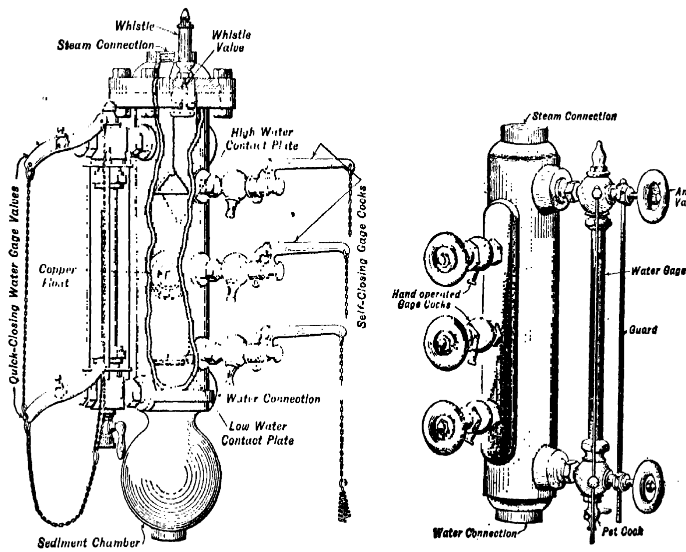


FIG. 44. — Water Columns.

tions. Mud and sediment are removed through a $\frac{3}{4}$-inch blow-off pipe connection attached to the lower end of the water column.

A **water gage,** Fig. 45, makes the water level visible from the boiler room floor. It is attached to the front of the water column, the lowest visible part of which must be 2 inches or more above the lowest permissible water level. The water gage consists of a strong glass tube which is connected to two needle valves by stuffing boxes and is protected by guards from being accidentally broken. The lower needle valve has a pet-cock connection used to clean the gage glass by blowing steam through it.

Because the pipes, which connect the water column to the boiler, may become clogged, or the stuffing boxes become leaky and the water gage thus indicate incorrectly, three valves, Fig. 46, called **gage cocks** are attached to the water column within the visible range of the water glass. The middle gage cock should be at the mean water level of the boiler, with the other two gage cocks located at equal distances above and below the middle cock, the distance varying from 3 to 5 inches according to the size of the boiler. The lower cock should be above the lowest safe water level. Vertical and cross-drum boilers often have the water column omit-

ted and the water gage and gage cocks attached directly to the shell as shown in Figs. 28 and 171.

A boiler may be provided with two water glasses located not less than 3 feet apart on the same level, in which case the gage cocks may be omitted.

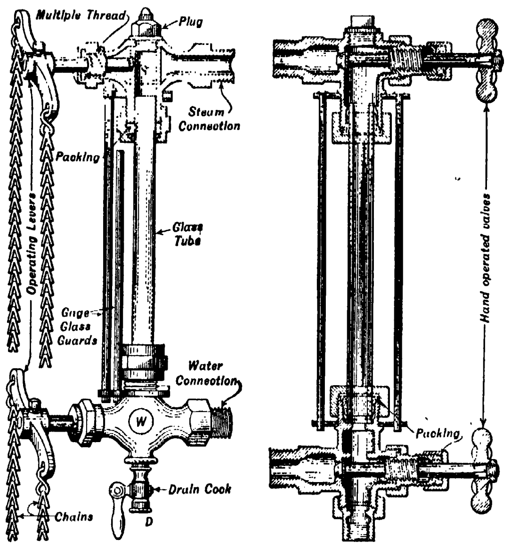


FIG. 45. — Water Gages.

The water column is sometimes provided with a **high and low water alarm**; a seamless copper float, inside the column, is so arranged that it will admit steam and thus blow a whistle, when the water level becomes too high or too low. Many engineers prefer not to use a high and low water alarm but to rely upon constant attention of the water tender.

39. Fusible Plug. — The fusible plug, Fig. 47, is used to protect the boiler against low water level. It consists of a bronze casing threaded on one end and having a conical hole from end to end, the taper of the hole being not less than $\frac{5}{8}$-inch to the foot. The hole must be reamed and tinned, before being filled, with an alloy having 99 per cent tin. The melting point

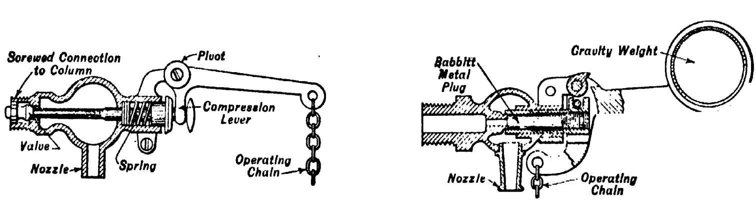


FIG. 46. — Gage Cocks.

of the alloy must be above the temperature corresponding to the steam pressure and below the temperature of the hot furnace gases. *The location of the fusible plug in any boiler must be such that the highest surface of the boiler exposed to heat of the gases will be protected from danger of over-*

heating. Water ordinarily covers the plug, and its temperature is not above that of the water. In case the water level should become low enough to uncover the plug, the alloy will melt and permit steam to escape, thus attracting the attention of the fireman.

The location of the fusible plug in a few typical boilers is as follows:

Horizontal return-tubular boiler: — In the rear head, not less than 2 inches above the upper row of tubes, and projecting through the sheet not less than 1 inch.

Locomotive type of boiler: — In the highest part of the crown sheet.

Vertical fire-tube boiler: — In an outside tube not less than one-third the length of the tube above the lower tube sheet.

Water-tube boiler, Heine type: — In the front course of the drum, not less than 6 inches above the bottom of the drum.

For location in other types of boilers, consult the A. S. M. E. BOILER CODE.

The fusible plug is not entirely reliable. The water end may become coated with scale from the water, or the gas end may become coated with incrustations from the gases. Both scale and incrustation are poor conductors of heat; the scale would cause the alloy to melt before it should, and the incrustation would prevent the alloy from melting when it should. In any case, the end exposed to the gases should be kept clean. *It must be replaced, at least once a year.*

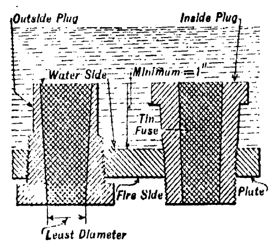


FIG. 47. — Outside and Inside Types of Fusible Plugs.

40. Safety Valves. — A safety valve is used to protect boilers against excessive pressure, by automatically discharging steam when the pressure rises above a definite point, at which the valve is set to open.

The safety valve should be bolted directly to the steam nozzle, without pipe, bends or valves. It should be large enough to discharge the maximum amount of steam that the boiler is capable of generating, without building up the discharge pressure more than 6 per cent. If a discharge pipe from the safety valve is used, it should be properly dripped and should have an open end. Each valve should have its own discharge pipe.

There are three principal types of safety valves, **the direct loaded, the lever and weight,** and **the direct spring loaded.** The first is not used to any extent in the United States. The lever-and-weight safety valve is simple in construction, but is not used for important work because it is liable to leak and can be easily tampered with. The direct spring-loaded valve is the only type permitted by the A. S. M. E. BOILER CODE.

Direct Spring-loaded Safety Valve. — An external and sectional view of a direct spring-loaded safety valve is shown in Fig. 48, with the various parts named. The body, cap and bonnet are made of cast iron or steel.

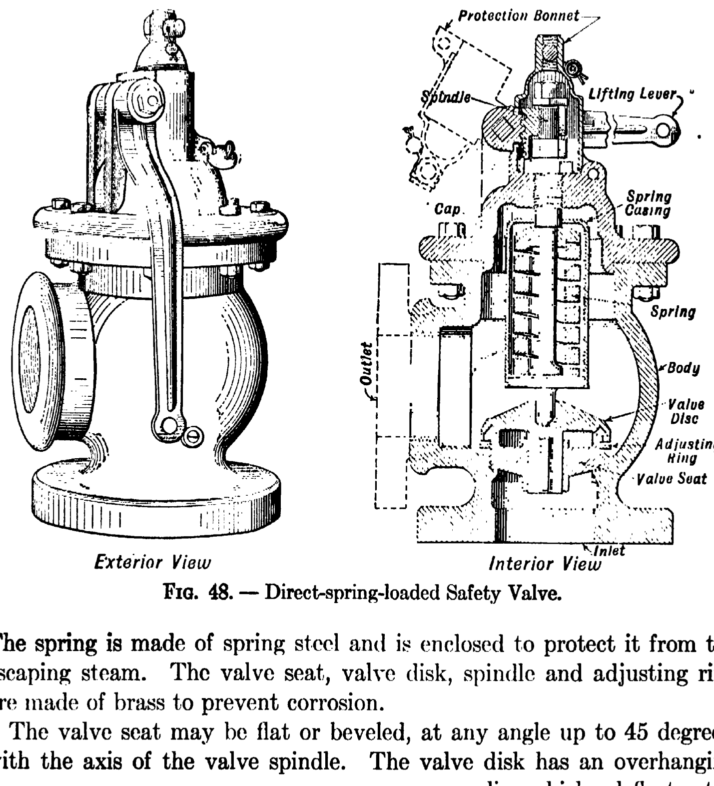


FIG. 48. — Direct-spring-loaded Safety Valve.

The spring is made of spring steel and is enclosed to protect it from the escaping steam. The valve seat, valve disk, spindle and adjusting ring are made of brass to prevent corrosion.

The valve seat may be flat or beveled, at any angle up to 45 degrees, with the axis of the valve spindle. The valve disk has an overhanging lip which deflects the escaping steam downward through holes in the outer edge of the seat. This lip, Fig. 49, affords additional area upon which steam may act when opening the valve. The valve disk also has wings which guide the valve so that it will seat squarely after opening.

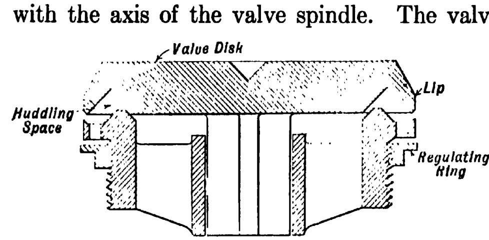


FIG. 49. — Valve Disk and Seat for Safety Valve.

A lifting lever is provided, so that the valve may be operated by hand when desired. It should be capable of lifting the valve $\frac{1}{16}$-inch from its

seat. The valve should be lifted from its seat once a day in order to be sure that it will operate properly. This may be done by hand, but it is better practice to raise the steam pressure until the valve opens, and then note the steam pressure at which it opens.

Operation of Pop, or Direct Spring-loaded, Safety Valve. — Steam pressure acts upon the valve disk and forces it slightly out of its seat. The steam, escaping into the space below the lip of the valve, finds its passage to the outer air obstructed. It has to pass out through the holes provided or below the outer edge of the lip of the disk. Pressure is built up in this **huddling space,** by the simmering of the valve, until the combined accumulated pressure and the reaction due to the steam that escapes downward cause the valve to open suddenly, or with a **pop.**

The position of the regulating ring determines the drop in pressure, sometimes called **blow down,** required before the valve is again seated. The amount of blow down usually varies from 3 to 8 pounds per square inch. The nearer the ring is to the holes in the valve seat, the greater must be the reduction in pressure before seating. The method of adjustment for " blow down " differs in the various makes of spring-loaded safety valves. The principle, however, is the same in all.

The adjustment for the pressure at which the valve will open is made by tightening or loosening the pressure of the spring upon the valve disk by means of an adjusting nut which is provided at the top of the spindle for this purpose. The spring pressure is generally applied below the seat of the valve, to prevent tipping of the valve.

Twin valves, that is, two spring loaded valves having a common Y base, are sometimes used in places where more than one valve is required, and a single connection to the boiler is desired.

The size or capacity of a spring-loaded safety valve can be determined by the formulae proposed in the A. S. M. E. BOILER CODE. The calculations may be based on the heat units in the fuel or on the amount of steam generated. In any case it should be capable of delivering all the steam the boiler can generate without having an excessive pressure built up within the boiler.

The discharge capacity in pounds of steam per hour may be found as follows:

$$W = 110 \times P \times D \times L \text{ for bevel seats at 45 degrees} \quad (1)$$
$$W = 155 \times P \times D \times L \text{ for flat seats} \quad (2)$$
$$W = 50 \times P \times A \quad \text{for seats at any angle} \quad (3)$$

in which W = weight of steam that a safety valve will handle, lb. per hr.
P = absolute boiler pressure, lb. per sq. in.
D = inside diameter of valve seat, in.

L = vertical lift of valve disk, measured with 3 per cent excess pressure, in.

A = relieving area in sq. in. = $3.1416 \times D \times L$ sine of seat angle.

Example 1. — Find the size of a "pop" safety valve required to discharge 10,150 lb. of steam per hour, if the valve rises from its seat a distance of 0.11 in. Seat is beveled at an angle of 45° and the absolute boiler pressure is 239.7 pounds per square inch.

Solution. — Rearranging Equation (1) and substituting the following values.

$W = 10{,}150$, $P = 239.7$ lb. per square inch, D = required diameter, $L = 0.11$ in.

$$D = \frac{W}{110 \times P \times L} = \frac{10{,}150}{110 \times 239.7 \times 0.11} = 3\tfrac{1}{2} \text{ in. diam.}$$

Locomotive Pop Safety Valve. — This valve is a spring type or safety valve with the spring entirely encased. As generally made, the entire valve, except the spring, is of brass and is provided with a muffler consisting of a plate or plates having a large number of small holes through which the escaping steam must pass. This diminishes the noise made by the rapid passage of the steam.

41. Feedwater Connections and Valves. — The feedwater pipe may enter the boiler through the front head or through the top of the front course, with the connection to the boiler made as shown in Fig. 17, page 25. The feed pipe line requires the following valves: a **check valve,** Fig. 50, which permits flow in one direction only and automatically prevents the back flow

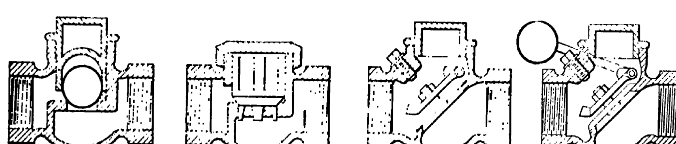


Fig. 50. — Types of Check Valves.

of water from the boiler when neither the feed pump nor the injector is working; a **valve** or **cock,** which is placed between the check valve and the boiler, and which ordinarily remains wide open except when it is desired to inspect or repair the check valve; and sometimes a **globe valve** (Art. 244, page 225) placed between the check valve and the source of water supply to regulate the amount of entering feedwater.

42. Bottom Blow-off Connection and Valves. — The blow-off connection to the boiler is made to the lowest water space practicable. The connection to the boiler should be by a screwed flange riveted to the shell, as shown in Fig. 18, page 25 . *The blow-off pipe and fittings should be extra strong and should be protected from the hot gases of the furnace, by means of a firebrick wall, by a substantial cast iron removable sleeve, or by a covering of non-conducting*

material. In spite of this protection the blow-off may burn out. To prevent this, the water in the blow-off pipe is often made to circulate, by means of a pipe which connects the water space of the boiler to the blow-off pipe outside the setting. The joint where the blow-off pipe passes through the setting should permit free expansion and contraction. If the pipe discharge is hidden from view, a tell-tale should be provided as a guard against leaks. *Outside the setting, a blow-off valve, Fig.* 51, *and a cock, Fig.* 52, *are required in the blow-off piping when the pressure is above* 125 *pounds per square inch.* When the pressure is below 125 pounds per square inch, either a blow-off valve or a cock may be used.

The blow-off cock must have the plug held in place by a guard or gland, and the upper end of the plug marked in line with the opening in the plug. A blow-off valve larger than $2\frac{1}{2}$ inches or less than 1 inch should not be used.

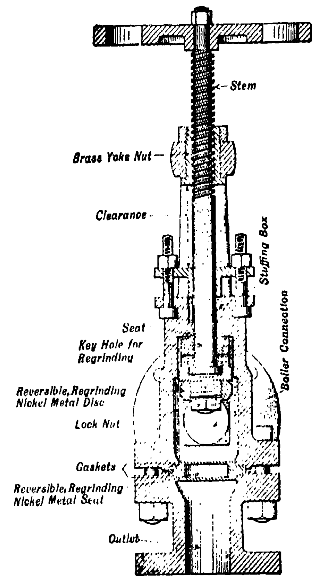


FIG. 51. — Blow-off Valve.

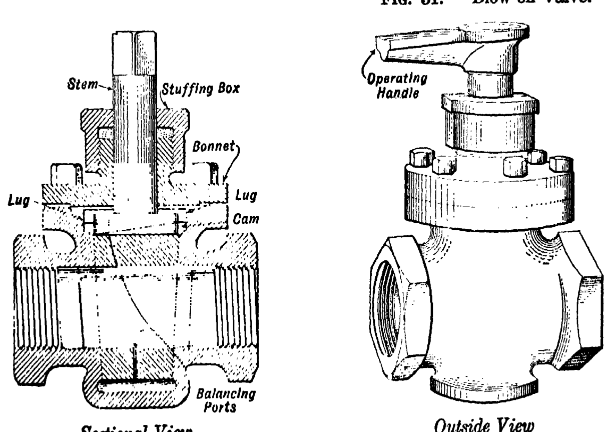


FIG. 52. — Blow-off Cock.

The use of a globe valve as a blow-off valve is prohibited.

43. Surface Blow-off. — To remove scum and other floating impurities, a surface blow-off, Fig. 53, is sometimes placed at the water level. The

inner and outer pipe must be screwed into a brass bushing in the boiler head in such a manner that it makes a smooth passage. The size of pipe used must not be larger than $1\frac{1}{2}$ inches.

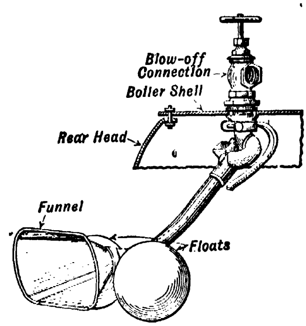


FIG. 53. — Surface Blow-off.

44. Blow-off Tank. — When the location of boilers is such that they cannot be blown down directly into the open, the discharge is made into a tank made of steel plate. The tank has a manhole, an open vent pipe, and inlet and outlet pipes connecting with the blow-off pipe and the sewer.

Sufficient water may be blown off at one time to fill the tank. The water is then allowed to cool and, when cool, is discharged into the sewer.

The discharge of hot water into the sewer is not permitted in most cities, because hot water disintegrates the tile sewer pipe and may cause trouble if under pressure.

45. Steam-pressure Gage. — Each boiler must have a **steam-pressure gage** connected to the steam space, or to the steam connection of the water column. The type of gage ordinarily used for indicating steam pressure is known as the Bourdon tube gage.

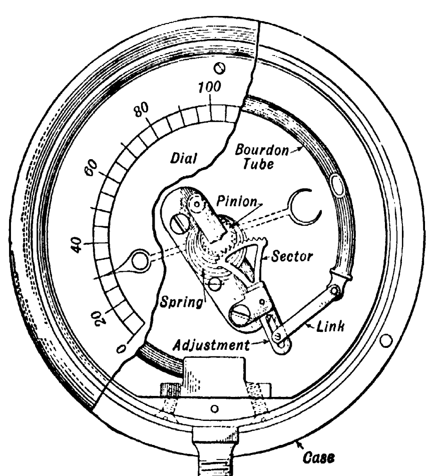


FIG. 54. — Bourdon-tube Steam Gage.

Bourdon Tube Steam Gage. — The Bourdon tube steam-pressure gage, with the dial removed, is shown in Fig. 54. Inside the case of the gage is a tempered copper or steel tube, or spring, of oval cross section, bent into an arc of a circle. One end of the tube passes out through the case to which it is attached, and is provided with a $\frac{1}{4}$-inch pipe thread for making the pressure connection. The other end is closed and is free to move under change in pressure. This free end of the tube is connected by a small link to an adjustable arm of a **toothed sector,** which moves about a pivot. The sector engages with a small pinion

mounted on the shaft to which the **pointer,** or **hand,** is attached. The hand moves over **a dial graduated in pounds per square inch.**

Pressure applied to the inside of the tube causes the section of the tube to become more nearly a true circle. This changes the radius of the arc to which the tube is bent, and moves the free end outward, thus rotating the shaft to which the pointer is fastened. A **hair spring** attached to the pinion shaft keeps the teeth of the sector and pinion in contact and compensates for lost motion.

Bourdon tube gages which are subjected to jar, such as locomotive gages, are made with the tube supported in the center and the free ends up. They are then called **double spring** gages. When the gage is thus supported the vibration of the needle caused by jarring is reduced to a minimum. Gages used on locomotives are protected externally from the heat of the boiler by wooden blocks placed between the gage and the frame to which it is attached.

A **water siphon,** Fig. 55, is used to prevent steam from coming into contact with the inside of the gage tube and thus destroying its accuracy. It consists of a chamber holding sufficient water to completely fill the tube.

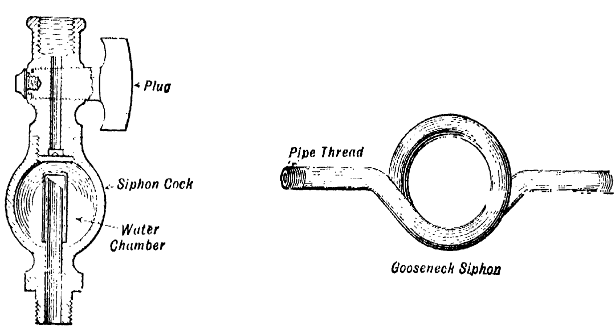


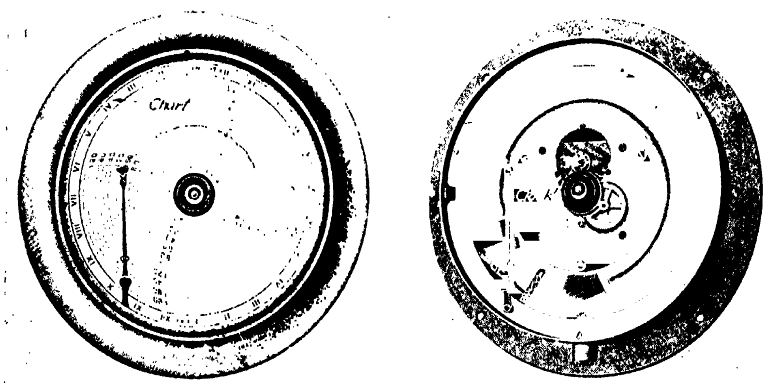
FIG. 55. — Gage Siphons.

It is placed between the gage and the boiler and should be of such a form that the water will not be easily drained from it. The pipe connecting the gage to the boiler should be of brass, and should have a T- or L-handled gage cock which will line up with the pipe connection when open. The cock should preferably be below the siphon to prevent the water escaping from the siphon if the cock should leak.

Diaphragm Gage. — This type of gage is not as common as the Bourdon tube gage, from which it differs in that a corrugated diaphragm, held between two flanges, replaces the Bourdon tube. The pressure acting on one side of the diaphragm deflects it, and the deflection is communicated

to the needle through a suitable system of levers. The deflection of the diaphragm is proportional to the pressure acting upon it.

Recording Gages. — To obtain a graphic record of the variation of pressure in the boiler, a recording gage, Fig. 56, is used. Its construction is similar to that of an ordinary steam gage, with the addition of a **clock**



External View *Internal View*

FIG. 56. — Pressure Recording Gage.

mechanism attached to the dial. The dial may make a complete revolution in twenty-four hours or some other period of time, depending on the type of clock used. A chart is attached to the movable dial, and a pen attached to the pointer of the gage records the variation in pressure upon the chart.

46. Calibration of Gages. — Gages are subject to variations in use and therefore require frequent calibration, or comparison with a standard. For this purpose a **dead-weight tester** or a **mercury column** is used. When using the dead-weight tester the gage is subjected to pressures produced by standard weights and the readings on the dial, as indicated by the hand of the gage, are compared with the pressures in pounds per square inch exerted by the standard weights. For many purposes, **standard test gages,** known to be correct by calibration with the dead-weight tester or mercury column, are used for comparision. The piping connections below a boiler gage are required to provide for the connection of a test gage.

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Cyclopedia of Engineering, Simmons-Boardman Co.

REVIEW QUESTIONS

1. Name the fundamental types of steam boilers considering the relative position of water and hot gases.

2. Define (*a*) boiler shell, (*b*) setting, (*c*) furnace, (*d*) water level, (*e*) heating surface.

3. State whether the following are water-tube or fire-tube boilers: (*a*) Locomotive, (*b*) Babcock and Wilcox, (*c*) Heine, (*d*) Wickes, (*e*) Stirling, (*f*) Dyson express.

4. Name three types of fireboxes used on locomotive boilers.

5. Name five kinds of bracing, or staying, used in boiler construction and state a possible application of each.

6. Give the names of eight setting fixtures and state the purpose of each.

7. Name the classes of water-tube boilers according to header construction.

8. At what point does the feedwater enter (*a*) a return tubular boiler, (*b*) horizontal water-tube boiler?

9. What type of boiler is the Scotch Marine boiler?

10. Name six boiler fittings and state the function of each.

CHAPTER III

PHYSICAL UNITS AND THEIR MEASUREMENT

47. Foreword. — All engineering depends upon the correct application of certain basic principles, which underlie physics, chemistry, mechanics, hydraulics, and thermodynamics. The fundamental principles and the units of measurement involved should be thoroughly understood, as reference will be made to them from time to time.

48. Matter. — Matter may exist in the solid, the liquid, or the gaseous state. In each of these states it is conceived to be made up of minute particles called **molecules,** which are further subdivided into smaller parts called **atoms.** According to the modern theory, atoms are made up of a number of extremely small particles called **electrons,** partly positive and partly negative in character. All matter consists ultimately of these electrons held together by mutual attraction.

The molecules existing in matter are considered as being in continuous motion and as exerting an attraction between themselves. This attraction is strongest in the solid, because, even though the molecules are in constant motion, the form of the solid remains unchanged. In the liquid state the attraction is less, and the molecules conform to the shape of the containing vessel. In the gaseous state the attraction appears to be still less powerful, and the gas or mixture of gases will always fill the vessel which contains it, whatever its size or shape.

Matter has **mass** and **inertia.** *A force is required to put it in motion, to change its direction of motion or to bring it to rest when in motion.* This property of matter is called **inertia.**

Matter may be changed from one form to another, but the total quantity remains unchanged; upon this fact is based the law known as the **Law of the Conservation of Matter.** It may be stated thus: *The total quantity of matter in the universe remains constant.*

49. Units of the F. P. S. System. — The system of units generally employed in engineering work, for measurement of matter and energy, is the **Foot Pound Second System,** often called for the sake of brevity, the **F. P. S.** system.

The **unit of time,** T, according to this system, is the second, or the $\frac{1}{86400}$ part of a mean solar day. Time is often expressed in minutes and hours.

The **unit of length,** L, is the foot. (= 0.3048 meter.)

The **unit of weight,** W, is the pound. (= 0.4536 kilogram.)

The **unit of area,** A, is the square foot or the square inch, as preferred.

The **unit of volume,** V, is the cubic foot. Volume equals the product of the cross-sectional area and the length. In calculations involving the quantity of air, Q is often used for the number of cubic feet.

The **unit of force** is the pound; centrifugal force is the force which a body exerts by reason of its rotation, and equals $\frac{Wv^2}{gr}$, in which W = weight in lb., v = velocity ft. per sec., g = 32.2 and r = radius in feet.

Pressure is the force acting on a body per unit of area.

The **density,** D, of a substance is the weight of a unit volume. The density of a few common substances is given in Table 2.

TABLE 2. — DENSITIES OF COMMON SUBSTANCES

	Substance	Specific Gravity	Weight per Cu. Ft., Lb.	Temp. °F.	Weight per Cu. In., Lb.
Liquids	Mercury	13.6	848.7	60	0.4906
	Water, max. density	1.00	62.43	39	0.036
	Water	0.958	59.83	212	
	Water, sea	1.02	64.0		
	Petroleum	0.87	54.0		
	Kerosene	0.78 to 0.82	50		0.0289
	Gasoline	0.70 to 0.75	46		
Solids	*Coal, anthracite	0.75 to 0.93	47 to 58		
	Coal, bituminous	0.64 to 0.87	40 to 54		
	Coal, coke	0.37 to 0.51	23 to 32		
	Coal, ashes		40 to 45		
	†Coal, anthracite	1.4 to 1.8	97		
	Coal, bituminous	1.2 to 1.5	84		
	Ice	0.88 to 0.92	56		
	Concrete	1.5 to 2.4	100 to 144		
	Sand or gravel		60		

* Coal as piled in bin. † Coal in solid form.

Speed is the rate of motion of a body, measured by the space passed over in a unit of time, and is usually expressed in feet per second. When equal distances are passed over in equal times the motion is **uniform** and when the distances are unequal the motion is **non-uniform.**

Velocity, v, differs from speed in that it involves the direction as well as the rate of motion. The velocity of a rotating particle may be expressed as: (1) **tangential,** or **linear velocity,** that is, the velocity in feet per second at which a point at a given radius r is traveling; or (2) **angular velocity,** that is, the number of unit angles per second through which the radius turns. The **unit angle,** or **radian,** is the angle subtended by an

arc equal to the radius. Thus, if N = number of revolutions per minute, and r = radius in feet

$$\text{Tangential velocity} = \frac{2\pi rN}{60} \quad \dots \dots \dots \dots \quad (4)$$

$$\text{Angular velocity} = \frac{2\pi rN}{60\,r} = \frac{\pi N}{30} \quad \dots \dots \quad (5)$$

Acceleration, a, is the rate of change of velocity and is expressed in feet per second per second, generally written "ft. per sec.^{2}." Acceleration may be either positive or negative. *It is positive if the velocity of the body is increasing, and negative if it is decreasing.* The acceleration of gravity, g, at sea level and at latitude 45 degrees is 32.174 ft. per sec.^{2}. It is generally taken as 32.2 ft. per sec.^{2}.

Mass, so called, is the weight of a body divided by the acceleration of gravity, or $\frac{W}{g}$. The **unit of mass** is a derived unit which equals the quantity of matter to which a unit force (1 lb.) will give an acceleration of 1 ft. per sec.^{2}. A pound, to the physicist, means one pound mass, whereas to an engineer it means a pound weight, or in other words, a force, since the weight of a body equals the force with which its mass is drawn toward the earth. The size of a unit mass is therefore the mass of a standard pound divided by 32.2.

50. Relation between Velocity, Acceleration, Time, Force and Mass. — For bodies starting from rest, the relation existing between velocity, acceleration, and time is written $V = at$. The force acting on a body is proportional to the acceleration produced; hence, the relation between force, mass and acceleration is written $F = Ma$, or substituting $\frac{W}{g}$ for M,

$$F = \frac{W}{g}a \quad \dots \dots \dots \dots \dots \quad (6)$$

in which F equals the force in pounds and the other units are as defined in Art. 49. Since acceleration equals the velocity divided by the time, Equation (6) may be written $F = \frac{Wv}{gt}$.

51. Energy. — Energy may be defined as the **ability to overcome resistance.** It exists in a great variety of forms, such as light, heat, sound and electricity. A body which has the ability to perform work is said to possess **mechanical energy,** which is measured in foot-pounds and may exist in either of the following forms,

1. Potential energy, or energy possessed by reason of position or deformation.
2. Kinetic energy, or energy possessed by reason of motion.

Potential and kinetic energy are interchangeable and when one form is converted into the other the amount of energy of the second form exactly equals

that of the first form. This fact is well illustrated by the pendulum of a clock; the pendulum at the top of its swing possesses potential energy; as it swings downward, its potential energy is given up to produce velocity; at the lowest point in the travel of the pendulum the potential energy has been converted into kinetic energy, and as the pendulum swings upward again the kinetic energy is converted into potential energy.

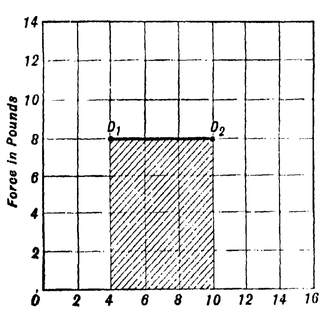
Kinetic energy is measured in foot-pounds and is calculated by the equation $K = \frac{Wv^2}{2g}$, in which K = ft-lb. of kinetic energy, W = weight of body in lb., v = velocity in ft. per sec., and g = acceleration of gravity = 32.2 ft. per sec.^{2}.

Electrical energy is usually measured in **joules.** A **joule** is the work done when one coulomb of electricity is conveyed between two points which differ in potential by one volt.

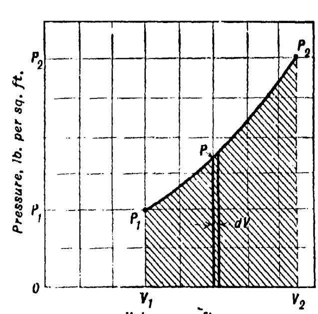
52. Work. — Work may be defined as the **overcoming of a resistance through space.** The unit of work is the quantity of energy expended by a force of one pound when acting through a distance of one foot in the line of action of the force. Unless there is motion, work is not performed in a mechanical sense. Work is expressed in foot-pounds and is calculated by the equation

Work in foot-pounds = Force in pounds × a distance in feet.

53. Work Diagram. — Work may be represented by a diagram, that is, an area bounded by lines representing force and the distance through



(*a*) *Force constant*



(*b*) *Force or pressure varying*

FIG. 57. — Work Diagrams.

which it is exerted, or by lines representing pressure and change in volume. *The area enclosed represents work and is generally expressed in foot-pounds.*

Force Constant. — The path traveled by a body, when under the action of a constant force of 8 pounds, may be represented graphically as in Fig. 57*a*. The work diagram, in this case, is a rectangle, and the work required to move the body from D_1 to D_2 equals the area 4 D_1 D_2 10

$$\text{Work} = (D_2 - D_1) \times 8 = 6 \times 8 = 48 \text{ ft-lb.}$$

Force Changing. — If the force or pressure varies, as represented by the line $P_1 P_2$, Fig. 57*b*, while the work is being done, the work area $V_1 P_1 P_2 V_2$ may be considered as made up of a series of very narrow sections of width (dV) for each of which there is a pressure P in pounds per square inch. The area $P\,dV$ represents the work (dW) done during the small change in volume dV. The summation of these elementary areas, dW, from V_1 to V_2 while the pressure is changing from P_1 to P_2 gives the work area $V_1 P_1 P_2 V_2$. When the pressures and volumes are such that $P_1 V_1 = P_2 V_2$, this area can be shown to be

$$P_1 V_1 \log_e \frac{V_2}{V_1} = \text{work in ft-lb.} \quad \ldots\ldots\ldots \quad (7)$$

in which $\log_e \frac{V_2}{V_1}$ = the Naperian logarithm of $\frac{V_2}{V_1} = 2.3 \log_{10} \frac{V_2}{V_1}$

When the pressures and volumes are such that $P_1 V_1^n = P_2 V_2^n$, the area can be shown to be

$$\frac{P_1 V_1 - P_2 V_2}{n - 1} = \text{work in ft-lb.} \quad \ldots\ldots\ldots \quad (8)$$

54. Power. — Power is the **rate of performing work,** and hence involves time as a factor. It is equal to the **amount of work performed** divided by the **time.** The unit of mechanical power is the **horsepower** (hp.) which equals 550 foot-pounds of work per second, or 33,000 foot-pounds per minute. The horsepower relation may be expressed:

$$\text{horsepower} = \frac{F \times v}{550} \quad \ldots\ldots\ldots\ldots \quad (9)$$

in which F = the force in pounds and v = velocity in feet per second. A mechanical unit of power exerted continuously for one hour is known as a **horsepower-hour.**

The unit of electrical power is the **watt,** which equals the product of the **volt times** the **ampere.** The **volt** is the unit of electrical pressure, or difference in potential, and the **ampere** the unit of electrical current. A watt is equivalent to one joule of work per second. The **kilowatt,** a larger unit of electrical power, is equal to 1000 watts. A kilowatt of power delivered continuously for one hour is called a **kilowatt-hour.**

55. Heat. — Heat is a form of energy. The heat of a body is the combined energy of the moving molecules of which every substance is composed. Heat is observed and recorded by its effect upon matter, producing change in shape, volume, and internal stress, change of state, as ice to water, change of temperature, and electrical and chemical effects. Conversely, heat may be obtained from mechanical and electrical energy, from chemical changes and from changes of physical state.

Heat which changes the temperature of a body is called **sensible heat.**

Its intensity may be measured by a thermometer. The heat that is used in changing the state of the body, as in changing ice into water or water into steam, is called **latent heat,** and while it is being added, no temperature change is indicated.

56. Temperature. — Temperature is a manifestation of the intensity of heat in a body, and is an indication of the rate of molecular activity. If a body is capable of transmitting heat to another body unaided, the first body is said to be at a higher temperature than the second body. When two bodies are at the same temperature, neither has any tendency to transmit heat to the other.

Temperature is ordinarily measured by instruments called thermometers and pyrometers, of which there are many forms. The **mercury thermometer,** which depends upon the uniform expansion and contraction of mercury to indicate temperature change, is most commonly used to measure temperature. It consists of a glass tube, or stem, with a small uniform bore, having its lower end enlarged to form a bulb. All air is removed from the stem, and the bulb and part of the stem are filled with mercury. The tube is marked in equal divisions, called degrees, which are numbered according to the scale of temperature to be employed. Since mercury expands when heated and contracts when cooled, the temperature of a body is obtained by placing the thermometer in contact with it and noting the height at which the mercury stands.

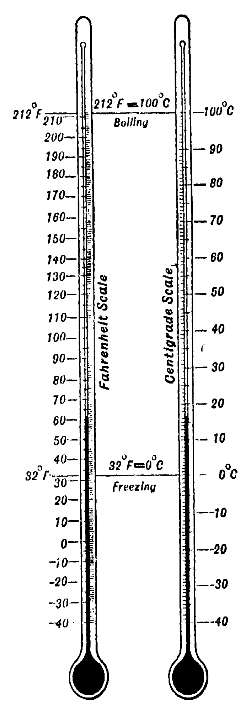


Fig. 58. — Thermometer Scales.

The unit of temperature measurement is the **degree.** It is capable of exact determination, provided two points can be obtained at which the intensity of heat is always constant. The melting-point of ice and the boiling-point of water, at atmospheric pressure (14.7 pounds per square inch), are the points usually selected. The thermometer is first placed in melting ice, and the height at which the liquid stands in the stem is marked. Then the thermometer is immersed in steam, from water boiling at atmospheric pressure, and the point reached by the top of the liquid is marked. The distance between these two points is divided into 180, 100 or 80 divisions, or degrees, according to whether the scale of temperature to be used is the Fahrenheit, Centigrade, or Réaumur. The last-named scale is seldom used at the present time.

The **Fahrenheit scale,** Fig. 58, largely used by engineers in America, takes the temperature of melting ice as 32 deg. fahr., and the temperature of boiling water, at atmospheric pressure, as 212 deg. fahr. The **Centigrade scale,** commonly used by scientists throughout the world, makes the melting point of ice 0 deg. cent., and the boiling point of water, at atmospheric pressure, 100 deg. cent. Since 100 deg. cent. equals 180 deg. fahr., one degree Fahrenheit equals five-ninths of a degree Centigrade, and one degree Centigrade equals nine-fifths of a degree Fahrenheit. The following equations may be used to convert temperatures from one scale to the other:

$$\text{Fahrenheit degrees} = \tfrac{9}{5}\ \text{Centigrade degrees} + 32 \quad . \quad . \quad . \quad (10)$$

$$\text{Centigrade degrees} = \tfrac{5}{9}\ (\text{Fahrenheit degrees} - 32) \quad . \quad . \quad . \quad (11)$$

Professor Sweet's rule for converting degrees Centigrade into degrees Fahrenheit is simple and easily applied. "*Double the number of degrees Centigrade, subtract one-tenth of this value, and add* 32."

The glass from which thermometers are made may undergo small changes from time to time. For this reason, it is necessary, in order to obtain accurate results, to **calibrate the thermometer,** by comparing it with a standard thermometer, and noting its variations. Besides calibrating the thermometer, it is necessary to make a correction for stem exposure when extreme accuracy is desired. This correction is given by the equation:

$$\text{Stem correction in degrees} = 0.000085\,N\ (t - t_s) \quad . \quad . \quad . \quad (12)$$

in which the decimal is the difference between the coefficient of expansion, or increase in length per inch per degree, of mercury and of glass, N = number of degrees of emergent mercury column, t = observed temperature and t_s = mean temperature of emergent stem

57. Absolute Temperature and Absolute Zero. — Besides the above temperature scales, there is another called the absolute scale of temperature. It is used in all calculations in which the temperature of gas volumes is involved. This scale is based on the so-called "absolute zero of temperature," or the point at which a perfect gas is considered to have zero volume. Scientists have found that a perfect gas, of which air is taken as a type, expands or contracts $\frac{1}{491.6}$ of its volume at 32 deg. fahr. for each deg. fahr. change in temperature. The absolute zero, therefore, may be taken as 491.6 deg. fahr. below the melting point of ice. This equals 459.6 degrees below zero on the Fahrenheit scale, or 273 degrees on the Centigrade scale. The absolute temperature (deg. fahr.) of a substance is found by adding 459.6 to the observed temperature. If the observed temperature were 60 deg. fahr. the absolute temperature would be 459.6 + 60, or 519.6 deg. fahr. The value 460 is used in ordinary engineering calculations, instead of 459.6.

58. Pyrometers. — Temperatures above 500 deg. fahr. are measured by **pyrometers.** High-grade thermometers, having nitrogen under pressure enclosed in the tube above the mercury, can be used for temperatures as high as 1000 deg. fahr. Such a thermometer is termed a mercurial pyrometer. The most common types of pyrometers are **expansion, thermo-electric, radiation, optical,** and **Seger cone.**

Metallic Pyrometer. — The metallic pyrometer consists essentially of two metal rods having widely different rates of expansion, such as iron and brass, and so connected as to move a pointer over a graduated scale during a change in temperature. Such pyrometers should not be used for temperatures above 1000 deg. fahr.

Thermo-electric Pyrometer. — This type of pyrometer, Fig. 59, comprises a thermocouple, an indicating or recording device, and suitable connecting wires. The thermocouple, Fig. 60, consists of two wires of dissimilar metals, and of different electrical

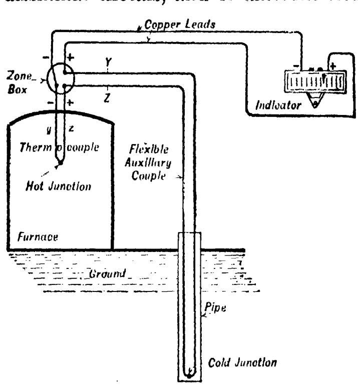


Fig. 59. — Thermo-electric Pyrometer.

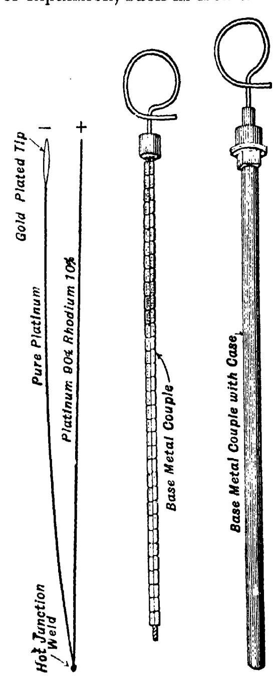


Fig. 60. — High Resistance and Base Metal Thermocouples.

conductivity, welded together at one end. When the weld or, "**hot junction,**" is heated and the other ends joined to form a "**cold junction,**" an electric current, which is proportional to the difference in temperature between the hot and cold junction, will flow. A **galvanometer,** an instrument for indicating small electric currents, is generally placed in the electric circuit, and the current is read from it. It is calibrated to read in degrees, by comparison with a standard thermometer. In actual operation, the "cold junction" is usually immersed in an ice bath or buried in the ground, in order to maintain it at a constant temperature and

the "hot junction," or furnace end, is placed in an iron or porcelain tube to protect it from breakage and deterioration.

Base metal thermocouples of low resistance are made of $\frac{1}{8}$-inch wires of nickel steel and copper, nickel steel and chromium or No. 8 gage iron and constantin, and are satisfactory for temperatures up to 1800 deg. fahr. For temperatures below 3000 deg. fahr., **high resistance thermocouples,** made with one wire of pure platinum and the other 90 per cent platinum and 10 per cent rhodium, are used.

Radiation Pyrometer. — Temperatures above 2500 deg. fahr. are measured by a radiation pyrometer, which consists of a cylindrical tube containing a concave mirror and a lens which is focused on the hot object. The mirror concentrates the rays upon a small thermo-electric couple connected to a galvanometer. The temperature reading is obtained in the same way as with a thermo-electric pyrometer.

Optical Pyrometer. — The measurement of temperature with this instrument is based on the fact that light varies in a definite manner with changes of temperature. Red light is separated from all the other light emitted by the incandescent body, and its intensity is compared with the intensity of light of the same color from a standard source of light, such as a special lamp having a tungsten filament. The eye is sensitive to differences in brightness between superimposed surfaces. The optical pyrometer, Fig. 61, varies the intensity of the standard light by changing the current supplied, until the filament appears of the same brightness as the hot object when viewed through the eye piece. When a balance has been obtained, the reading of the milli-ammeter is made and the corresponding temperature is read from a curve supplied with the instrument.

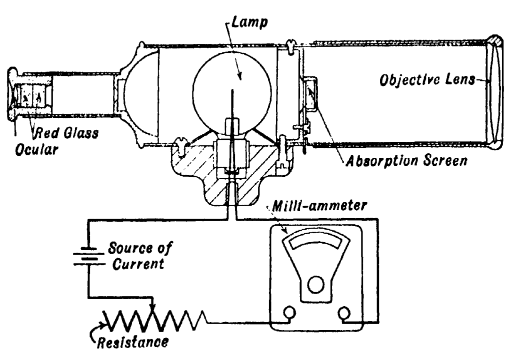


FIG. 61. — Optical Pyrometer.

Seger Cone Pyrometer. — Furnace temperatures may be obtained by the use of metals having different melting-points. The oxides of the metals are made into cones which are graded to melt at temperatures differing by 100 to 200 deg. fahr. Several cones are placed in the furnace, and the temperature is nearest that corresponding to the melting temperature of the cone, the top of which has just bent sufficiently to touch the plate upon

which the cones rest. Seger cones can be used for temperatures from 500 to 1900 deg. fahr.

Accuracy of Pyrometers. — The mercurial pyrometer is the most accurate for low-temperature measurements. The electrical pyrometer is the best for high-temperature measurements. Expansion pyrometers are subject to wide variations and should only be used after careful calibration. On high-temperature measurements the deviation from accuracy of a pyrometer may be as high as 40 deg. fahr.

59. Quantity of Heat. — Heat may be expressed by the usual energy units, such as foot-pounds or joules. It is the custom in engineering work to express the quantity of heat by a separate unit known as the **British thermal unit,** B.t.u., which is **the amount of heat required to raise the temperature of one pound of pure water one degree Fahrenheit,** often taken from 62 deg. fahr. to 63 deg. fahr. The **mean,** or **average,** B.t.u. is most commonly used in engineering calculations. It is the average amount of heat per degree required to raise the temperature of one pound of water from 32 deg. fahr. to 212 deg. fahr.

60. Specific Heat. — *The specific heat of a substance is the quantity of heat necessary to raise the temperature of one pound of the material one degree.* It varies with the physical properties of the substances and with the temperature of the substance. Its numerical value is obtained by comparison with water as a standard. Two specific heats are recognized:

1. The "**true**" **specific heat** measured at the temperature stated.
2. The "**mean**" **specific heat** which is an average over the temperature range considered.

The mean specific heat may be expressed by an equation as follows:

$$C = \frac{H}{W\,(t_2 - t_1)} \quad \ldots\ldots\ldots\ldots \quad (13)$$

in which C = an average specific heat over the temperature range from t_1 to t_2.

H = quantity of heat required to raise W pounds of substance from t_1 to t_2.

In dealing with gases, a further distinction is made between the specific heat at constant pressure, C_p, and the specific heat at constant volume, C_v, as explained in Art. 69, page 80.

61. Relation between Heat, Work and Energy. — It has been shown by physicists that *mechanical energy and heat are mutually interchangeable.* This is known as the **First Law of Thermodynamics.** The relation which exists between heat and mechanical energy was first determined by Joule, and is known as **Joule's equivalent.** The most recent value states that 1 B.t.u. = 777.64 foot-pounds. The value 778 is generally used for engineering calculations. Expressed mathematically:

$$1 \text{ B.t.u.} = 778 \text{ foot-pounds} \quad \ldots\ldots\ldots \quad (14)$$

$$1 \text{ foot-pound} = \frac{1}{778} \text{ B.t.u.} \quad \ldots\ldots\ldots \quad (15)$$

62. Relation between Heat, and Electrical and Mechanical Power. — The unit of mechanical power is the horsepower, which is equal to 2547 B.t.u. per hour, or 33,000 × 60 = 1,980,000 foot-pounds per hour.

The unit of electrical power is the kilowatt and is equal to 3414 B.t.u. per hour. The following relations therefore exist and should be memorized, as they are of much use in engineering calculations:

1 kilowatt-hour	= 3414 B.t.u.
1 horsepower-hour	= 2547 B.t.u.
1 horsepower-hour	= 0.746 kilowatt-hour
1 kilowatt	= 1.34 horsepower

63. Heat Transmission. — There are three methods of transmitting heat; namely, conduction, convection and radiation.

Conduction is a molecular transmission of heat through the substance, from molecule to molecule. Such transmission will take place between any two parts of a substance which are at different temperatures.

All substances conduct heat, though the rate at which heat is conducted varies with the material. Substances which transmit heat readily are called good conductors, and those which transmit heat slowly, poor conductors. The heat transmitted by conduction varies with the thickness of the material, the area in contact and the temperature difference.

Convection is the transmission of heat by circulation of a fluid or gas over the surface of the hotter or colder body. This circulation may be due to natural causes or it may be produced mechanically. The quantity of heat transferred by convection does not depend upon the nature of the material or its absolute temperature. It does depend on the velocity of the moving fluid or gas, the form and dimensions of the body, and the temperature difference between the moving substance and the contact surface of the body.

Radiation is the transmission of heat through an agency commonly known as ether which is assumed to occupy all intermolecular space. It takes place in straight lines and obeys the same laws as light. The rate at which radiant heat is emitted or absorbed depends upon the character of the surface of the hot or cold body, the temperature difference between the surface and surroundings, and the absolute temperature. It does not depend upon the form of body, however, unless there are re-entrant surfaces to intercept the rays.

64. Measurement of Pressure. — The unit of pressure is usually the *pound per square inch* and is written "lb. per sq. in." Low pressures are often expressed in inches of mercury or inches of water. Pressure is ordinarily indicated by a gage, Fig. 62. Gages indicating pressures above

that of the atmosphere are called **pressure gages** and those showing pressures below atmospheric pressure are called **vacuum gages.**

The simplest type of pressure-measuring device is the manometer which is a glass tube bent into the form of a U and partially filled with water or mercury, Fig. 63*a*. One side, A, of the tube is attached to the vessel in which the pressure is to be measured, the other side, B, is left open to the atmosphere. With equal pressures on each leg of the U, the surfaces of the liquid remain stationary. Upon the application of pressure to leg A, the surface of the liquid will fall in leg A, and rise in leg B. The distance between the level of the liquid in A and B, multiplied by the weight in pounds per cubic inch of the liquid, gives the difference in pressure between A and B, expressed in pounds per square inch.

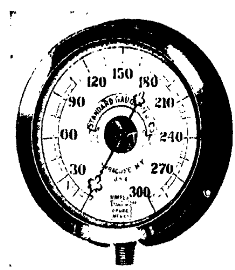


FIG. 62.—External View of Pressure Gage.

Manometers using water should be read at the bottom of the meniscus and those using mercury should be read at the top of the meniscus, to make the error from capillarity as small as possible. *Manometers are only used for low pressures.*

Manometers used for measuring the difference in pressure between the inside and outside of a chimney or furnace are called **draft gages,** a common

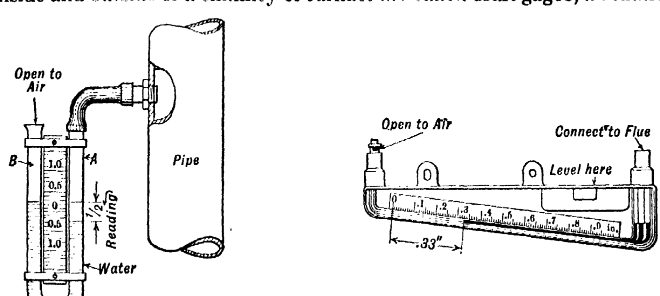
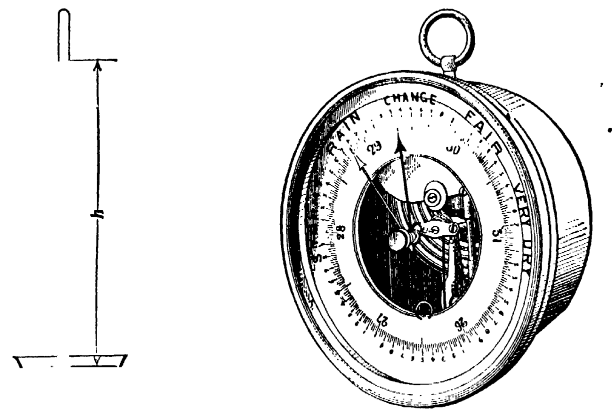


FIG. 63. — Manometer and Draft Gage.

form being shown in Fig. 63*b*. If one leg of the U tube is arranged on an incline, the distance moved by the liquid is increased for a given pressure change. The inclined leg ordinarily rises 1 inch in 10. A light oil is used in the gage to reduce the effect of capillarity and to give a greater deflection than can be obtained by the use of water. The gage is calibrated to read equivalent inches of water, by comparison with a gage using water, and can easily be read to 0.001 inch.

65. Barometers. — The pressure of the atmosphere is measured by means of barometers, of which there are two principal types, the mercurial barometer, Fig. 64*a*, and the aneroid barometer, Fig. 64*b*. The latter is



(*a*) *Simple Barometer* (*b*) *Aneroid Barometer*

FIG. 64. — Barometers.

similar to a delicate pressure gage and is calibrated by comparison with the mercurial barometer.

The Mercurial Barometer. — One type of this barometer, shown in Fig. 64*a*, consists of a glass tube about 3 feet long sealed at one end. After all air has been excluded, the tube is filled with mercury and is inverted in a bath of mercury. The mercury in the tube then drops to a certain point, where it remains. This height, *h*, represents the pressure of the atmosphere in inches of mercury and varies with the altitude above sea level.

A commercial, or standard, type of this instrument, shown in Fig. 65, consists of a glass tube closed at the top and enclosed in a brass tube having a movable scale at the top. The lower end of the glass tube dips into a glass cup having a leather bottom resting on a movable disk. The level of the mercury in the cup can be raised and lowered by means of an adjusting screw until its surface touches a fixed ivory point which is located at the zero of the measuring scale.

Standard atmospheric pressure is defined as the pressure of a column of pure mercury, 29.92 inches high, at a temperature of 32 deg. fahr. Expressed in pounds per square inch, it is 14.7. To correct observed barometer readings to standard conditions, see method of correction, Art. 522, page 533.

66. Absolute Pressure and Vacuum. — Pressure gages show the difference between the pressure in a vessel and the pressure of the atmosphere. *Absolute pressure is the pressure above the zero of pressure.* It is found

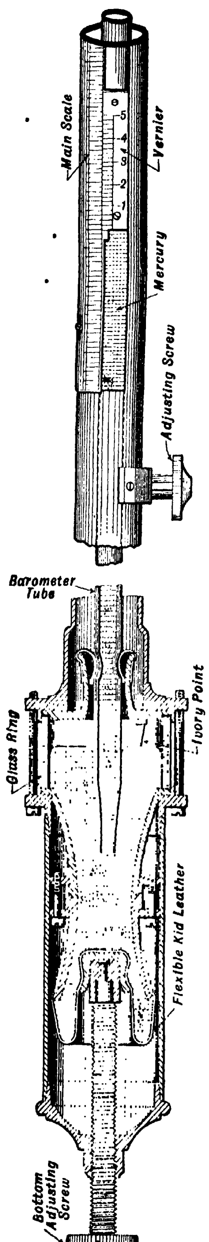


FIG. 65. — United States Weather Bureau Barometer.

by adding the pressure shown by the barometer to the pressure shown by the gage, both expressed in the same units.

Absolute pressure = gage pressure + atmospheric pressure.

Vacuum may be defined as the absence of pressure, a vessel having no pressure within it would have an absolute vacuum. As ordinarily used in engineering, the word vacuum means only a partial vacuum. For instance, a vacuum of 18 inches means that the pressure corresponds to 18 inches of mercury below the pressure of the atmosphere; if the barometer reading were 30 inches of mercury, the absolute pressure would be (30 − 18), or 12 inches of mercury absolute.

67. Conversion of Pressure. — It is frequently necessary to convert inches of mercury or water to pounds per square inch. Since the weight of a cubic inch of mercury at 70 deg. fahr. is 0.4906 pound, and of water at the same temperature is 0.0360 pound, pressure in inches of mercury can be converted to pounds per square inch by multiplying by 0.491, and pressures in inches of water can be converted to pounds per square inch by multiplying by 0.036.

The pressure expressed in feet of water existing on a system is ordinarily known as **head.** When so expressed, the pressure in feet may be converted to pounds per square inch by multiplying by 0.434. The factor 0.434 is obtained as follows: one cubic foot of water weighs 62.3 pounds at 70 deg. fahr., hence, a column of water one foot high, at 70 deg. fahr., and one square inch in cross section, will exert a pressure of $\frac{62.3}{144}$ or 0.434 pounds. To convert pounds per square inch to feet of water, multiply by the reciprocal of 0.434, or 2.31. *This value varies with the temperature of the water.*

Water or air flowing in a pipe has a velocity produced by the pressure existing between the two points between which flow takes place. The relation existing between the velocity and the pressure, in this case, is expressed by the equation $v = \sqrt{2gh}$,

in which v = velocity in feet per second, h = head in feet of fluid flowing and g = 32.2. This equation has a direct application in pump problems. The factor, h, is known as the **velocity head.**

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PROBLEMS AND REVIEW QUESTIONS

1. The side of a square coal bin is 14 ft. and its height is 16 ft. Express its volume in cubic meters.

2. One cubic foot of solid anthracite coal weighs 97 lb. Express this weight in kilograms.

3. How many more cubic feet of space will be required to store 1,000 tons of bituminous coal than to store the same amount of anthracite? Their respective weights, as placed in a bin, are 52.8 lb. and 53.4 lb. per cubic foot.

4. A locomotive is traveling west at 20 miles per hour. Find its velocity in feet per minute.

5. A locomotive exerts a pull of 45 tons on a train, when traveling at 5 miles per hour. What is the horsepower delivered by the locomotive?

6. An electric car starting from rest attains a speed of 30 miles per hour in five minutes Find the acceleration.

7. A pound of steam, at 20 lb. per square inch absolute pressure, has a volume of 20.10 cubic feet. What is its density?

8. A body weighs 190 lb. when weighed on a platform scale. Find its mass (so-called).

9. A revolving body weighs 200 lb. and has a linear velocity of 2,000 ft. per min. Find the kinetic energy of the body.

10. Define work and express as an equation. A compound duplex steam pump lifts 432,000 gallons of water, at 70 deg. fahr. through 300 feet, in 24 hr. A gallon of water contains 231 cubic inches. Find the work done per minute. (For density of water consult Table 5, page 85.)

11. Draw a diagram representing the work done by a constant force F acting through a distance d. Explain how the area of the diagram represents the work done by the force.

12 An air compressor compresses 300 cubic feet of air per min. from a pressure of 14.35 lb. per sq. in. abs. to a pressure of 100 lb. per sq. in., gage. Find the work done, using Equation (7).

13. A loaded box car weighs 60,000 lb. Find the force required to give it an acceleration of 80 ft. per minute, per minute.

14. The temperature of water entering a feedwater heater is 70 deg. fahr. Express as degrees Centigrade.

15. The temperature of water leaving a condenser is 50 deg. cent. Express as degrees Fahrenheit, using Professor Sweet's rule.

16. Express as absolute temperature, using the Fahrenheit scale, 30 deg. cent., 90 deg. fahr., — 10 deg. fahr.

17. Name five types of pyrometers and explain the method of measuring temperature, using one of the types named.

18. The water in a tank weighs 200 lb. The temperature of the water is changed from 60 deg. fahr. to 212 deg. fahr. Express the quantity of heat required in B.t.u. Mean specific heat of water taken as 1.00.

19. The mean specific heat of iron between 82 deg. and 2000 deg. fahr., is 0.126. Find the quantity of heat in B.t.u. given up, if 4 pounds of iron at 2000 deg. fahr. are placed in sufficient water to lower the temperature of the iron to 80 deg. fahr. Express this quantity of heat as ft-lb.

20. A mechanical horsepower is equal to 2547 B.t.u. per hour. Show how this value is obtained.

21. During the year 1920, a total of 32,739,000,000 kilowatt-hours of energy were delivered by the Central Stations of the United States. Express this as horsepower-hours.

22. A Corliss engine delivers 24 horsepower continuously for six hours. How many horsepower-hours is it delivering?

23. The steam gage attached to a boiler read 210 lb. per sq. in. The barometer reading was 29.92 in. of mercury. What is the absolute gage pressure in lb. per sq. in.?

24. The pressure of water in a water pipe is 2000 lb. per sq. in. What "head" of water would be required to produce this pressure if the water temperature is 65 deg. fahr.?

25. A vessel is under a pressure of 3 ft. of water at 120 deg. fahr. The density of water at 120 deg. fahr. is 61. 71 lb. per cu. ft. Find the pressure in lb. per sq. in. caused by the water.

26. The reading of a draft gage at the base of a stack in a power house is 1 inch of water. What pressure does this represent in lb. per sq. in., if the temperature of the air surrounding the gage is 70 deg. fahr.?

27. The velocity of water flowing in a water pipe is 200 ft. per minute. Find the pressure in lb. per sq. in. required to produce this velocity.

28. The factor for converting pressure in lb. per sq. in. into feet of water is 2.31 at 70 deg. fahr. Explain how this factor is obtained.

29. The permissible rise in temperature of the windings of an electric generator is given as 40 deg. cent. what is the corresponding temperature rise in deg. fahr.?

CHAPTER IV

PROPERTIES OF AIR, WATER AND STEAM

68. Foreword. — Some knowledge of the properties of air, water and steam is essential to an understanding of the operation of such power plant equipment as fans, compressors, chimneys, engines and condensers. Only sufficient material is here given to make clear the following chapters of the book.

69. Air. —Pure air is a mechanical mixture of oxygen and nitrogen in the following proportions:

	Per cent by Volume	*Per cent by Weight*
Oxygen..........	20.91	23.15
Nitrogen.........	79.09	76.85

As ordinarily found, air is not pure, but contains impurities, such as carbon dioxide, ozone, water vapor, and dust.

The **specific density,** or weight per cubic foot, of air decreases with an increase in temperature, and the specific volume therefore increases with the temperature. The density and volume of air at various temperatures are given in Table 3.

TABLE 3. — VOLUME AND WEIGHT OF AIR AT VARIOUS TEMPERATURES.

Barometer 29.92 Inches Mercury.

Temperature ° F.	Volume Cu. Ft. per Lb.	Weight Lb. per Cu. Ft.	Temperature ° F.	Volume Cu. Ft. per Lb.	Weight Lb. per Cu. Ft.	Temperature ° F.	Volume Cu. Ft. per Lb.	Weight Lb. per Cu. Ft.
32	12.39	.0807	160	15.62	.0640	340	20.15	.0496
50	12.84	.0779	170	15.87	.0630	360	20.66	.0484
55	12.97	.0771	180	16.12	.0620	380	21.16	.0473
60	13.10	.0763	190	16.37	.0611	400	21.66	.0462
65	13.22	.0756	200	16.62	.0602	425	22.29	.0449
70	13.35	.0749	210	16.88	.0593	450	22.92	.0436
75	13.47	.0742	212	16.93	.0591	475	23.55	.0424
80	13.59	.0735	220	17.13	.0584	500	24.18	.0414
85	13.72	.0729	230	17.38	.0575	525	24.81	.0403
90	13.85	.0722	240	17.63	.0567	550	25.44	.0393
95	13.98	.0715	250	17.88	.0559	575	26.07	.0384
100	14.10	.0709	260	18.14	.0551	600	26.70	.0374
110	14.36	.0697	270	18.39	.0543	650	27.96	.0358
120	14.61	.0685	280	18.64	.0537	700	29.22	.0342
130	14.86	.0673	290	18.89	.0529	750	30.48	.0328
140	15.11	.0662	300	19.14	.0522	800	31.74	.0315
150	15.36	.0651	320	19.65	.0509	850	33.00	.0303

Air has a specific heat at constant pressure, C_p, and a specific heat at constant volume, C_v. The former is the quantity of heat required to raise

the temperature of a unit weight of the gas one degree Fahrenheit at constant pressure. It varies from 0.2375 to 0.2430, but for most engineering calculations 0.24 is sufficiently accurate. *A gas expanding at constant pressure performs a certain amount of external work, and the heat equivalent of this work is included in the value of the specific heat at constant pressure.* The **specific heat at constant volume** is the quantity of heat required to raise the temperature of a unit weight of a gas one degree Fahrenheit, the volume remaining constant. As the gas does not change in volume, external work is not done, and hence, *the specific heat at constant volume is less than the specific heat at constant pressure by the amount of work done in the first case.* This fact is sometimes written, $C_p - C_v = R$, in which R is the amount of external work done, all quantities being expressed in the same units.

The specific heat at constant pressure for a mixture of gases is obtained by multiplying the specific heat of each constituent gas by the percentage weight of the gas in the mixture, adding the results and dividing the sum by 100.

70. Laws of Perfect Gases and Their Application to Air. — Boyle's, or **Mariotte's Law** refers to the relation existing between the pressure and volume of a gas when the temperature remains constant. The law is stated as follows: *when the temperature is constant, the volume of a given weight of a gas varies inversely as the absolute pressure.* Stated as an equation:

$$\frac{V_1}{V_2} = \frac{P_2}{P_1}, \qquad \text{or} \qquad P_1V_1 = P_2V_2 = \text{a constant} \quad . \quad . \quad . \quad (16)$$

in which P_1 = initial absolute pressure, lb. per. sq. ft.
P_2 = final absolute pressure, lb. per sq. ft.
V_1 = initial volume of a given weight of gas, cu. ft.
V_2 = final volume of a given weight of gas, cu. ft.

When a change occurs in a gas, such that the product of pressure and volume is constant with the temperature also remaining constant, the change is called **isothermal.**

Example 2. — Two hundred cubic feet of air at atmospheric pressure is compressed at constant temperature until the final pressure is 80 lb. per sq. in., gage. What volume will it occupy?

Solution. — Using Equation (16) and making proper substitutions:

$$P_1V_1 = P_2V_2, \qquad \text{or} \qquad V_2 = V_1\frac{P_1}{P_2} = \frac{200 \times 14.7 \times 144}{(80 + 14.7) \times 144} = 31 \text{ cu. ft.}$$

$P_1 = 14.7 \times 144$; $P_2 = (80 + 14.7) \times 144$; $V_1 = 200$ cu. ft.; V_2 = required.

Charles', or **Gay-Lussac's Law** refers to the relation between pressure, volume and temperature of a gas when either the pressure or the volume remains constant during the change in absolute temperature. It may be stated thus: *upon the addition of heat to a gas, the volume of a given weight varies directly as its absolute temperature when the pressure remains constant;*

or the pressure of a given weight of a gas varies directly as the absolute temperature when the volume remains constant. Stated as an equation:

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}, \quad \text{or} \quad \frac{V_1}{V_2} = \frac{T_1}{T_2} \quad \ldots\ldots\ldots \quad (17)$$

in which T_1 = initial absolute temperature, deg. fahr.

T_2 = final absolute temperature, deg. fahr., and other symbols as given in Equation (16).

Example 3. — One pound of air at sea level and atmospheric pressure has a volume of 12.387 cubic feet when the temperature is 32 deg. fahr. Find its volume at 62 deg. fahr. and atmospheric pressure.

Solution. — Using Equation (17) and making proper substitutions:

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}, \quad \text{or} \quad V_2 = V_1\frac{T_2}{T_1} = \frac{522 \times 12.387}{492} = 13.14 \text{ cu. ft.}$$

V_2 = required; V_1 = 12.387 cu. ft.

$T_1 = (32 + 460) = 492$ deg. fahr. abs.; $T_2 = (62 + 460) = 522$ deg. fahr. abs.

71. The Combined Law of Gases. — When the pressure, volume and temperature, P_1, V_1 and T_1, of a quantity of air or gas are all changed, the resulting values P_2, V_2 and T_2 may be found by the combined laws of Boyle and Charles', giving the equation:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \text{a constant} \quad \ldots\ldots\ldots \quad (18)$$

Considering, for convenience, that this action takes place in two steps:

1. With the absolute temperature T_1 constant, let the pressure be changed to P_2 according to Boyle's Law. The resulting volume V_n can then be obtained as follows:

$$\frac{P_1}{P_2} = \frac{V_n}{V_1}, \quad \text{and} \quad V_n = V_1\frac{P_1}{P_2} \quad \ldots\ldots \quad (19)$$

2. From the condition P_2, V_n, T_1, let the change be made to a final temperature T_2, according to Charles' Law.

Then
$$\frac{V_n}{V_2} = \frac{T_1}{T_2}, \quad \text{and} \quad V_n = V_2\frac{T_1}{T_2} \quad \ldots\ldots \quad (20)$$

Equating the values of V_n from Equations (19) and (20):

$$\frac{P_1V_1}{P_2} = \frac{T_1V_2}{T_2}, \quad \text{or} \quad \frac{P_1V_1}{T_1} = \frac{P_nV_n}{T_n} = \text{a constant}$$

If the constant per pound of air or gas is called R, the resulting equation for one pound of gas may be expressed $PV = RT$, and for any weight W of a gas

$$PV = WRT \quad \ldots\ldots\ldots\ldots \quad (21)$$

in which P = absolute pressure, lb. per sq. ft.

V = volume, cu. ft.

T = absolute temperature, deg. fahr.

R = constant for any given gas. Values of R are given in Table 4

The numerical value of R is the number of foot-pounds of external work done when one pound of gas is raised in temperature one degree at constant pressure, and it may be obtained by calculation using: (1) the international atomic weights, with air as a base; (2) the constant volume and constant pressure specific heats of the gas, as $R = 778\ (C_p - C_v)$; or (3) the observed volumes under standard conditions, as

$$R = \frac{\text{Volume of 1 lb. of a gas} \times \text{pressure in lb. per sq. ft.}}{\text{Absolute temperature}} = \frac{VP}{T}$$

For application of the first two methods, the reader is referred to any of the standard books on thermodynamics, while the third method is illustrated by *Example* 4.

Example 4. — The volume occupied by one pound of air at a pressure of 14.69 lb. per sq. in. abs. is 12.39 cu. ft., when the temperature is 32 deg. fahr. Find the value of R for air.

Solution. — Using the third method previously explained:

$$R \text{ for one pound of air } = \frac{VP}{T} = \frac{12.39 \times 14.69 \times 144}{460 + 32} = 53.37.$$

Example 5. — For use in calculating the results of an air compressor test, it is desired to know the density, or weight per cubic foot, of air at 70 deg. fahr. and atmospheric pressure.

Solution. — From the combined law of gases, Equation (21),

$$V = \frac{WRT}{P} = \frac{1 \times 53.37 \times 530}{14.7 \times 144} = 13.35 \text{ cu. ft.}$$

$$\text{Density} = \frac{1}{V} = \frac{1}{13.35} = 0.0749 \text{ lb. per cu. ft.}$$

W = weight of gas in lb. = 1; R = 53.37, ft.-lb.
T = absolute temperature = (70 + 460) = 530 deg. fahr.
P = absolute pressure in lb. per sq. ft. = 144 × 14.7

TABLE 4. — THERMAL AND PHYSICAL PROPERTIES OF COMMON GASES

Gas	Molecular Chemical Symbol	Specific Gravity Air = 1	Weight per Cu. Ft. Lb.*	Specific Heat: Constant Pressure C_p	Specific Heat: Constant Volume C_v	$n = \frac{C_p}{C_v}$	R
Air	....	1.000	0.0807	.2375	.1689	1.406	53.37
Oxygen	O_2	1.053	.0892	.2175	.1551	1.402	48.55
Nitrogen	N_2	0.967	.0783	.2438	.1727	1.412	55.32
Hydrogen	H_2	0.069	.0056	3.409	2.412	1.413	775.66
Carbon Dioxide	CO_2	1.529	.1227	.2169	.167	1.299	38.82
Carbon Monoxide	CO	0.967	.0781	.2450	.174	1.408	55.24
Methane	CH_4	0.558	.0447	.5930	.450	1.320	96.31
Ethylene	C_2H_4	0.967	.0780	.4040	....	....	....
Ethane	C_2H_6	1.075	.0838	....	....	....	....
Acetylene	C_2H_2	0.920	.0725	.350	.270	1.280	59.37
Sulphur Dioxide	SO_2	2.264	.1786	.154	.123	1.250	24.10

* At 32 deg. fahr. and atmospheric pressure of 14.7 lb. per sq. in.

72. Work at Constant Pressure. — The work done when a gas expands at constant pressure is equal to the pressure in pounds per square foot, multiplied by the change in volume in cubic feet. As an example, consider the original volume of a gas as represented by the point a in Fig. 66, and the final volume, after the addition of heat, by point b. The work performed is represented by the shaded area and equals:

$$\text{Work} = P\,(V_2 - V_1) \quad \ldots\ldots\ldots \quad (22)$$

in which P = pressure on the gas, lb. per sq. ft.
V_2 = larger volume of gas, cu. ft.
V_1 = smaller volume of gas, cu. ft.

73. Water. — Pure water is a chemical combination of two elements, hydrogen and oxygen. These elements when combining chemically always do so in the proportion of two parts, by volume, of hydrogen to one part, by volume, of oxygen. If the two gases are mixed while cold in the above proportions, the mixture is purely a mechanical one *until, by the influence of* heat, they combine chemically. If the union of the two gases is brought about in a vessel so arranged that the resulting water is maintained at a high temperature, it will retain its gaseous condition and will form two volumes of steam. Conversely, two volumes of steam may be dissociated by the application of heat into its constituent elements, two volumes of hydrogen and one volume of oxygen.

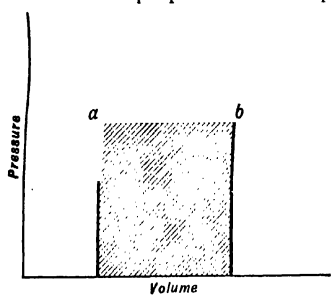


FIG. 66. — Diagram illustrating Work at Constant Pressure.

Water has been universally adopted as the standard by which the relative weights of other liquids and solids are determined, this relation being expressed by the term **specific gravity.** The specific gravity of any body indicates its weight as compared with the weight of an equal volume of pure water. The volume and weight, per cubic foot, of water changes with the temperature, as shown in Table 5. Its density is 62.427 pounds per cubic foot at 39.2 deg. fahr., the point of maximum density.

A United States standard gallon has a volume of 231 cubic inches, and a weight of $8\frac{1}{3}$ pounds at 62 deg. fahr.

Water is but slightly compressible, and for all practical purposes may be considered non-compressible. Its boiling point varies with the pressure; the higher the pressure, the higher the boiling point. See Art. 75, page 86.

The specific heat of water varies with the temperature, as shown in Table 6.

TABLE 5.—VOLUME AND WEIGHT OF WATER AT VARIOUS TEMPERATURES

(GOODENOUGH)

Temperature ° F.	Volume Cu. Ft. per Lb.	Weight Lb. per Cu. Ft.	Temperature ° F.	Volume Cu. Ft. per Lb.	Weight Lb. per Cu. Ft.	Temperature ° F.	Volume Cu. Ft. per Lb.	Weight Lb. per Cu. Ft.
32	0.01602	62.42	210	0.01670	59.88	390	0.0186	53.84
39.2	.01602	62.43	212	.01672	59.83	400	.0187	53.42
40	.01602	62.43	220	.01677	59.63	410	.0189	52.99
50	.01602	62.42	230	.01684	59.37	420	.0190	52.55
60	0.01603	62.37	240	0.01692	59.11	430	0.0192	52.11
70	.01605	62.30	250	.01700	58.83	440	.0194	51.66
80	.01607	62.22	260	.01708	58.55	450	.0195	51.20
90	.01610	62.11	270	.01716	58.26	460	.0197	50.70
100	.01613	62.00	280	.01725	57.96	470	.0199	50.20
110	0.01616	61.87	290	0.01735	57.65	480	0.0201	49.70
120	.01620	61.71	300	.01745	57.32	490	.0203	49.20
130	.01625	61.55	310	.01755	56.98	500	.0205	48.70
140	.01629	61.38	320	.01766	56.62	510	.0208	48.20
150	.01634	61.20	330	.01778	56.24	520	.0210	47.60
160	0.01639	61.00	340	0.01790	55.85	530	0.0212	47.10
170	.01645	60.80	350	.01803	55.46	540	.0215	46.50
180	.01651	60.58	360	.01816	55.06	550	.0218	45.90
190	.01657	60.36	370	.01829	54.66	560	.0221	45.20
200	.01663	60.12	380	.01843	54.24	600	.0235	42.60

Because of this variation in specific heat, the heat required to raise one pound of water through a given temperature range, known as the **heat of liquid,** will depend upon the mean value of the specific heat over this range. For ordinary calculations the specific heat of water may be taken as unity.

TABLE 6.—SPECIFIC HEAT OF WATER

Temperature ° F.	Specific Heat
30	1.0098
55	1.0000
100	0.9967
160	1.0002
210	1.0050

Water as found in nature is never pure, being always more or less contaminated by impurities, which often have serious effects when the water is fed into boilers. The impurities most commonly found in water are earthy matter, bi-carbonates of lime and magnesia, iron, sulphate of lime, chlorides and sulphates of magnesia, carbonate of soda in large amounts, acids, dissolved carbonic acid and oxygen, grease and organic matter. *Impurities are commonly reported as the number of parts per one thousand, or one hun-*

dred thousand. A larger proportion than 100 parts total solids per 100,000 parts of water should in most cases condemn the water for use in steam boilers. For a discussion of the effects of these impurities and methods of removing them, consult Chapter XII, page 230.

74. Steam. — All substances exist in a solid, a liquid or a gaseous form, depending upon the pressure and temperature to which they are subjected. For instance, water under an atmospheric pressure of 14.7 pounds per square inch absolute, is a solid for temperatures below 32 deg. fahr., a liquid for temperatures between 32 and 212 deg. fahr., and a vapor at 212 deg. fahr.; upon further rise in temperature it approaches a gaseous state. *Steam is water vapor* and as such does not obey the laws of perfect gases.* It exists as a vapor because sufficient heat has been added to the water from which it is formed to supply the latent heat of evaporation and change the liquid to a vapor. *This change takes place at a definite and constant temperature which depends solely upon the pressure under which the change takes place.*

75. Formation of Steam. — The effect of heat, when applied to water in a boiler is best explained by considering one pound of water at 32 deg. fahr. as enclosed in a vertical cylinder having a cross-sectional area of 1 square inch. The pressure of the atmosphere at sea level may, for convenience, be considered as replaced by a frictionless piston weighing 14.7 pounds per square inch.

If heat is now applied to the cylinder, the rate of vibration of the molecules of the water will be increased, and the resulting increase in kinetic energy will be shown by a rise in temperature of 1 deg. fahr. for each B.t.u added to the water, considering the specific heat of water as unity. This increase in temperature will continue until a point is reached at which the attraction between the molecules as a liquid is broken down and boiling, or evaporation, takes place. The point at which this occurs is called the **boiling-point,** and the amount of heat added between 32 deg. fahr. and the boiling point is known as the **heat of liquid.**

When the boiling point is reached, bubbles of steam form at the point of application of heat, and, being lighter than water, rise to the surface and discharge the steam which they contain. The temperature at which boiling takes place depends entirely upon the pressure. In the case being considered, boiling occurs at 212 deg. fahr.; while if the pressure were increased to 200 pounds per square inch absolute, boiling would occur at 381.9 deg. fahr., and if the pressure were lowered to, say, one pound per square inch absolute, the water would boil at 101.8 deg. fahr. *The temperature of the*

* When substances are changed from a liquid or solid to the gaseous state, the gas first formed does not obey the laws of perfect gases and when in this condition is called a *vapor*.

steam formed is in each case the same as that of the water from which it is formed. Steam at the temperature of evaporation is said to be **saturated.**

After the boiling point is reached, the temperature remains constant until the entire pound of water is evaporated. During this time, the heat that is added changes the physical state of the water, and the quantity of heat required to bring about the change is called the latent heat of evaporation. While evaporation is taking place, the volume of the cylinder occupied by water and steam is increased from the volume occupied by the water alone to that occupied by a pound of saturated steam, thus lifting the piston and performing work. After the entire pound of water has been converted into steam, the addition of more heat to the steam will increase the temperature above the temperature of evaporation and the steam will become **superheated.** At the same time, the volume is increased above that of saturated steam.

The changes taking place during the formation of steam may be illustrated by a diagram, as in Fig. 67, in which the line AB represents the

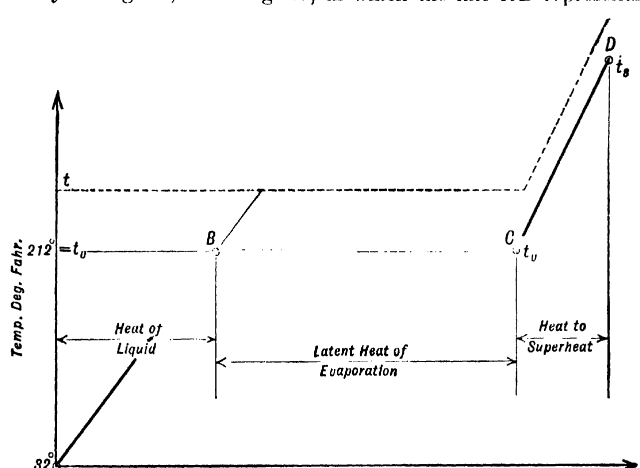


FIG. 67. — Diagram showing Relation between Temperature and Heat Added during Formation of a Pound of Steam.

change in the heat of liquid with the change in temperature; BC represents by a horizontal line the latent heat of evaporation and CD the change in the heat required to superheat the steam to some temperature above that of evaporation. The dotted line shows how an increase in pressure would effect the various quantities.

76. Dry and Saturated Steam. — In the previous article, *saturated steam was defined as steam at the temperature of evaporation corresponding to the pressure.* Saturated steam may be wet or dry, depending on the condi-

tions accompanying evaporation. *A pound of steam existing at the temperature and pressure of evaporation is dry and saturated if all the water from which it is formed has been evaporated;* when in this condition steam is invisible. If the entire pound of water from which the steam is formed is not evaporated, the steam will be **wet**; that is, will contain small particles of water in suspension which have not been evaporated.

77. Steam Tables. — To solve problems relating to equipment using steam, it is necessary to know the exact amount of heat supplied per pound of steam used; in addition, the volume of the steam is often desired. The various quantities mentioned in Art. 75 are called the **properties of steam,** and are tabulated in **Steam Tables,** arranged to give values for either saturated or superheated steam. The steam tables are based upon the following considerations: (1) all quantities of heat are given per pound of dry saturated steam, or per pound of superheated steam, as the case may be; (2) all heat quantities are stated in British thermal units above 32 deg. fahr.; and (3) all pressures are expressed as absolute pressure in pounds per square inch when above atmospheric pressure, and in pounds per square inch or inches of mercury when below atmospheric pressure.

78. Properties of Steam. — The numerical values of the properties of steam are the results of experimental data obtained by scientists, and compiled by themselves or by others who have studied their data. A thorough understanding of the use of the steam tables is necessary in order to make the calculations involved in the heat interchanges taking place in all apparatus using steam. The following properties of steam are those given in the **saturated steam table,** on the following page:

p = steam pressure in lb. per sq. in. abs.

t = temperature in degrees Fahrenheit corresponding to the absolute pressure.

v = specific volume of dry steam, cu. ft. per lb.

$\frac{1}{v}$ = density in lb. per cu. ft.

h = heat of liquid, B.t.u. above 32 deg. fahr.

L = latent heat of evaporation, B.t.u. per lb. of dry steam.

H = total heat, B.t.u. per lb. of dry steam.

ρ = internal latent heat of evaporation, B.t.u. per lb.

E* = internal energy of steam, B.t.u. per lb.

θ = entropy of water.

$\frac{L}{T}$ = entropy of evaporation.

ϕ = entropy of steam.

σ = volume of one pound of water, cu. ft. (See Table 5, for numerical values.)

* The numerical value of E is not given in the steam table on page 89. It may be obtained by adding the internal latent heat, ρ, to the heat of liquid, h.

TABLE 7.—PROPERTIES OF DRY AND SATURATED STEAM

(G. A. GOODENOUGH)

* Pressure		Temperature ° F.	Volume Cu. Ft. per Lb.	Heat of Liquid B.t.u.	Latent Heat in B.t.u.		Total Heat of Steam B.t.u.	Entropy		
In. of Mercury	Lb. per Sq. In.				of Evaporation	Internal		of Water	of Evaporation	of Steam
*	p	t	v or s	h or q	L or r	ρ	$H=L+h$	θ	L/T	ϕ
† •			v''	i'	r	ρ	i''	s'	r/T	s''
0.2	0.0982	34.55	2992.0	2.57	1071.7	1016.3	1074.2	0.0052	2.1687	2.1739
2	0.982	101.17	338.9	69.16	1036.0	974.3	1105.1	0.1316	1.8474	1.9790
4	1.965	125.44	176.5	93.37	1022.5	958.3	1115.9	0.1739	1.7478	1.9217
[illegible]	2.947	140.80	120.7	108.69	1013.9	948.1	1122.6	0.1998	1.6888	1.8886
8	3.929	152.26	92.1	120.2	1007.4	940.4	1127.5	0.2187	1.6464	1.8651
10	4.912	161.50	74.8	129.4	1002.1	934.1	1131.4	0.2336	1.6134	1.8556
12	5.894	169.30	63.0	137.2	997.5	928.8	1134.7	0.2461	1.5862	1.8323
14	6.88	176.06	54.6	143.9	993.6	924.1	1137.5	0.2568	1.5630	1.8198
16	7.86	182.06	48.14	149.9	990.0	920.0	1140.0	0.2662	1.5429	1.8091
18	8.84	187.46	43.12	155.4	986.7	916.2	1142.1	0.2746	1.5250	1.7996
20	9.82	192.38	39.08	160.3	983.8	912.7	1144.1	0.2822	1.5086	1.7912
22	10.81	196.89	35.75	164.8	981.1	909.6	1145.9	0.2892	1.4944	1.7835
24	11.79	201.09	32.95	169.0	978.5	906.6	1147.5	0.2955	1.4810	1.7766
26	12.77	205.00	30.57	173.0	976.1	903.8	1149.1	0.3015	1.4687	1.7702
28	13.75	208.67	28.53	176.6	973.8	901.2	1150.5	0.3070	1.4572	1.7642
29.92	**14.697**	**212.0**	26.81	**180.0**	**971.7**	898.8	1151.7	0.3120	1.4469	1.7589
	16	216.3	24.76	184.3	969.1	895.8	1153.4	0.3184	1.4337	1.7521
	17	219.4	23.40	187.5	967.1	893.5	1154.6	0.3230	1.4242	1.7473
	18	222.4	22.18	190.5	965.2	891.4	1155.7	0.3274	1.4153	1.7427
	19	225.2	21.09	193.3	963.4	889.3	1156.7	0.3316	1.4068	1.7384
	20	228.0	20.10	196.0	961.7	887.3	1157.7	0.3356	1.3987	1.7343
	22	233.1	18.38	201.2	958.4	883.6	1159.6	0.3430	1.3837	1.7267
	24	237.8	16.95	206.0	955.3	880.1	1161.3	0.3499	1.3698	1.7197
	26	242.2	15.73	210.4	952.4	876.8	1162.8	0.3563	1.3570	1.7133
	28	246.4	14.67	214.6	949.7	873.7	1164.3	0.3622	1.3452	1.7074
	30	250.3	13.76	218.6	947.1	870.7	1165.7	0.3679	1.3340	1.7019
	32	254.0	12.95	222.4	944.6	867.9	1166.9	0.3731	1.3236	1.6967
	34	257.6	12.24	225.9	942.2	865.2	1168.1	0.3781	1.3137	1.6918
	36	260.9	11.60	229.4	939.9	862.7	1169.2	0.3829	1.3044	1.6873
	38	264.2	11.03	232.6	937.7	860.2	1170.3	0.3874	1.2956	1.6830
	40	267.2	10.51	235.8	935.5	857.8	1171.3	0.3917	1.2871	1.6788
	42	270.2	10.04	238.8	933.5	855.5	1172.2	0.3958	1.2791	1.6749
	44	273.0	9.61	241.7	931.5	853.3	1173.2	0.3998	1.2714	1.6712
	46	275.8	9.22	244.5	929.6	851.2	1174.0	0.4036	1.2640	1.6676
	48	278.4	8.86	247.2	927.7	849.1	1174.8	0.4072	1.2570	1.6642
	50	281.0	8.53	249.8	925.9	847.1	1175.6	0.4108	1.2501	1.6609
	52	283.5	8.22	252.3	924.1	845.1	1176.4	0.4142	1.2436	1.6577
	54	285.9	7.93	254.7	922.4	843.2	1177.1	0.4174	1.2373	1.6547
	56	288.2	7.67	257.1	920.7	841.4	1177.8	0.4206	1.2311	1.6517
	58	290.5	7.42	259.5	919.0	839.5	1178.5	0.4237	1.2252	1.6489
	60	292.7	7.18	261.7	917.4	837.8	1179.1	0.4267	1.2195	1.6462
	62	294.9	6.97	263.9	915.8	836.0	1179.7	0.4296	1.2139	1.6435
	64	296.9	6.76	266.1	914.3	834.3	1180.3	0.4324	1.2085	1.6409
	66	299.0	6.57	268.2	912.7	832.7	1180.9	0.4352	1.2032	1.6384
	68	301.0	6.39	270.2	911.2	831.1	1181.5	0.4379	1.1981	1.6360

* Symbols used by Marks and Davis. † Symbols used by Goodenough.

TABLE 7. — (*Concluded.*) PROPERTIES OF DRY AND SATURATED STEAM (G. A. GOODENOUGH)

Pressure		Temperature °F.	Volume Cu. Ft. per Lb.	Heat of Liquid B.t.u.	Latent Heat in B.t.u.		Total Heat of Steam B.t.u.	Entropy		
In. of Mercury	Lb. per Sq. In.				of Evaporation	Internal		of Water	of Evaporation	of Steam
*	p	t	v or s	h or q	L or r	ρ	$H=L+h$	θ	$L/$[illegible]	ϕ
†			v''	i'	r	ρ	i''	s'	$r/$[illegible]	s''
	70	302.9	6.22	272.2	909.8	829.5	1182.0	0.4405	[illegible]	[illegible]
	72	304.8	6.05	274.2	908.8	827.9	1182.5	0.4431	[illegible]	[illegible]
	74	306.7	5.90	276.1	906.9	826.4	1183.0	0.4456	1.1835	[illegible]
	76	308.5	5.75	278.0	905.5	824.9	1183.5	0.4480	1.1789	[illegible]
	78	310.3	5.61	279.8	904.2	823.4	1184.0	0.4504	1.1744	[illegible]
	80	312.0	5.48	281.6	902.8	821.9	1184.4	0.4527	1.1700	[illegible]
	82	313.7	5.35	283.4	901.5	820.5	1184.9	0.4550	1.1657	[illegible]
	84	315.4	5.23	285.1	900.2	819.1	1185.3	0.4572	1.1615	[illegible]
	86	317.1	5.12	286.8	898.9	817.7	1185.7	0.4594	1.1574	1.6168
	88	318.7	5.01	288.5	897.7	816.3	1186.1	0.4615	1.1534	1.6149
	90	320.3	4.905	290.1	896.4	815.0	1186.5	0.4636	1.1495	1.6131
	92	321.8	4.805	291.7	895.2	813.7	1186.9	0.4657	1.1456	1.6113
	94	323.3	4.709	293.3	894.0	812.4	1187.3	0.4677	1.1419	1.6096
	96	324.8	4.617	294.8	892.8	811.1	1187.7	0.4697	1.1381	1.6079
	98	326.3	4.528	296.4	891.7	809.8	1188.0	0.4777	1.1345	1.6062
	100	327.8	4.42	297.9	890.5	808.6	1188.4	0.4736	1.1309	1.6045
	105	331.4	4.24	301.6	887.6	805.5	1189.2	0.4782	1.1222	1.6004
	110	334.8	4.06	305.1	884.8	802.6	1190.0	0.4827	1.1138	1.5965
	115	338.1	3.89	308.6	882.1	799.7	1190.7	0.4870	1.1058	1.5928
	120	341.3	3.74	311.9	879.5	796.9	1191.4	0.4911	1.0982	1.5893
	125	344.4	3.59	315.1	876.9	794.2	1192.0	0.4950	1.0908	1.5858
	130	347.4	3.46	318.2	874.4	791.6	1192.6	0.4989	1.0836	1.5825
	135	350.3	3.34	321.2	872.0	789.0	1193.2	0.5026	1.0767	1.5793
	140	353.1	3.23	324.2	869.6	786.4	1193.7	0.5062	1.0700	1.5762
	145	355.8	3.12	327.0	867.2	784.0	1194.2	0.5097	1.0636	1.5733
	150	358.5	3.02	329.8	864.9	781.6	1194.7	0.5131	1.0573	1.5704
	155	361.1	2.93	332.5	862.7	779.2	1195.2	0.5164	1.0512	1.5676
	160	363.6	2.84	335.2	860.5	776.9	1195.7	0.5196	1.0453	1.5649
	165	366.1	2.76	337.8	858.3	774.6	1196.1	0.5227	1.0395	1.5622
	170	368.5	2.68	340.3	856.2	772.4	1196.5	0.5258	1.0339	1.5597
	175	370.8	2.61	342.8	854.1	770.2	1196.9	0.5287	1.0284	1.5572
	180	373.1	2.54	345.2	852.0	768.0	1197.2	0.5316	1.0231	1.5547
	185	375.4	2.47	347.6	849.9	765.9	1197.6	0.5344	1.0179	1.5523
	190	377.6	2.41	350.0	847.9	763.9	1197.9	0.5372	1.0128	1.5500
	195	379.7	2.35	352.2	846.0	761.8	1198.2	0.5399	1.0079	1.5478
	200	381.9	2.29	354.5	844.0	759.8	1198.5	0.5426	1.0030	1.5456
	300	417.5	1.55	392.4	809.4	724.7	1201.9	0.5863	0.9229	1.5092
	400	448.8	1.16	422.0	780.6	695.9	1202.5	0.6190	0.8631	1.4821
	500	467.2	0.93	446.6	755.0	670.9	1201.7	0.6455	0.8146	1.4601
	600	486.5	0.77	468.0	731.8	648.5	1199.8	0.6679	0.7735	1.4414
	700	503.4	0.66	487.1	710.3	627.9	1197.4	0.6874	0.7376	1.4250
	800	518.5	0.57	504.3	690.1	608.8	1194.4	0.7048	0.7056	1.4104
	900	532.3	0.50	520.3	670.8	590.8	1191.1	0.7205	0.6764	1.3969
	1000	544.9	0.45	535.2	652.4	573.6	1187.6	0.7349	0.6496	1.3845
	1200	567.7	0.36	562.3	617.6	541.8	1179.7	0.7607	0.6015	1.3622

* Symbols used by Marks and Davis. † Symbols used by Goodenough.

79. [illegible] **ure and Temperature.** — The absolute pressure is the quantity upon [illegible] other properties depend; consequently, it is the first item in th[illegible] The temperature of evaporation, t_v, is the second item in t[illegible] s so closely related to the pressure. Under low pressure [illegible] increases rapidly for small increase in pressure, while ur[illegible] re the increase is less rapid — a fact which is of great im[illegible] peration of certain apparatus using steam.

80 [illegible] **id.** — *The heat of liquid, h, is the amount of heat added* [illegible] *temperature of a pound of water from 32 deg. fahr. to the boiling* [illegible] nd as given in the steam table is calculated from the equation

$$h = C_p \, (t_v - 32) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

[illegible] ich C_p is the mean specific heat of water or $\frac{1}{180}$ th. of the heat required o raise one pound of water from 32 to 212 deg. fahr., and t_v is the temperature of evaporation. For some calculations it is sufficiently accurate o consider C_p equal to 1; then the heat of liquid would equal

$$h = t_v - 32$$

However, for accurate work, the values as given in the steam table should be used. It must be remembered that the heat added is measured from 32 deg. fahr., and in case the temperature is other than 32 deg. fahr. the steam table value must be corrected as explained in Art. 89, page 94.

81. Latent Heat of Evaporation. — *The heat added to convert one pound of water into steam under a constant pressure is known as the latent heat of evaporation and is designated by L.* This heat is potential, since it is stored in the steam. Consequently upon condensation of the steam, the same amount of heat will be given up as was originally added during evaporation. At a pressure of 14.7 pounds per square inch the latent heat of evaporation is 971.7 B.t.u. per pound of steam.*

82. Internal and External Latent Heat. — The latent heat of evaporation is considered as composed of the internal latent heat, ρ, and the heat equivalent of the external work APu, in which $A = \frac{1}{778}$, P = the absolute pressure in pounds per square foot, and u equals the change in volume between one pound of water† and one pound of steam at the given pressure, as illustrated in Fig. 68. The internal heat is considered as used in changing the rate and amplitude of vibration of the molecules. The external latent heat is the heat equivalent of the work done in changing the volume from the volume of one pound of water, σ, to the volume of one pound of dry steam, v, the pressure remaining constant during the change.

* Marks and Davis' Steam Tables give 970.4 for L.

† For volume of one pound of water at various temperatures, see Table 5, page 85.

83. Total Heat per Pound of Dry Saturated Steam. — The total heat per pound of dry saturated steam is represented by the symbol H. *It is the total heat required to convert a pound of water at 32 deg. fahr. into dry steam,* and equals the sum of the heat of liquid, the internal latent heat and the external latent heat. In Fig. 67, page 87 it is shown by the horizontal distance from A to C and when written as an equation, is as follows:

$$H = h + \rho + 144\ Apu, \text{ or } H = h + L \quad . \ . \ . \ . \ . \ . \quad (24)$$

If the pressure under which evaporation takes place is increased, the temperature of evaporation is increased, the heat of liquid is increased and the latent heat of evaporation decreased. The net result, however, is to increase the total heat per pound of dry steam. (See Fig. 67, page 87.)

Example 6. — Steam is formed in a boiler in which the gage pressure is 115.3 lb. per sq. in. with the barometer reading 29.92 in. mercury. Find the volume, heat of liquid, internal latent heat and the total heat in each pound of dry steam formed.

Solution. — Referring to the Steam Table, page 89, the desired numerical values of the properties are found under the proper column and on a horizontal line corresponding to the absolute pressure in the boiler.

Atmospheric pressure = 29.92 × 0.491 = 14.7 lb. per sq. in.
Absolute pressure = 115.3 + 14.7 = 130 lb. per sq. in.
v = 3.46 cu. ft.; h = 318.2 B.t.u.; ρ = 791.6; H = 1192.6 B.t.u.

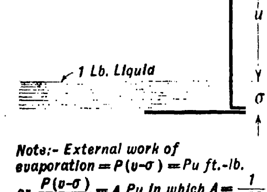


FIG. 68. — Diagram illustrating APu.

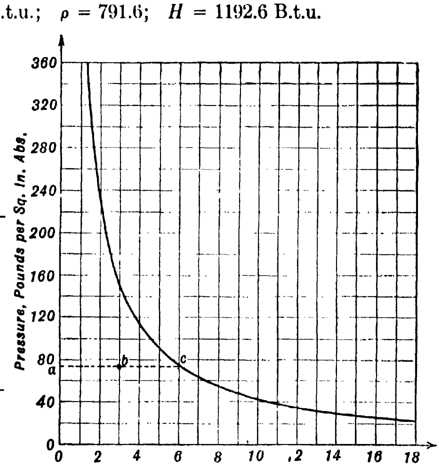


FIG. 69. — Curve showing Variation of Specific Volume with Change in Pressure.

84. Interpolation from Tables. — The finding of values which lie between those given in the steam tables is known as interpolation. In interpolating, the direction in which the quantities vary must be kept in mind, in order to make the correction in the proper manner.

Example 7. — Find the latent heat per pound of dry and saturated steam when the absolute steam pressure is 91.7 lb. per sq. in.

Solution. — From Steam Table, page 89, it is seen that

L for a pressure of 90 = 896.4 B.t.u.
L for a pressure of 92 = 895.2 B.t.u.
Difference = 1.2

Of this difference, $1.7 \times 1.2 \div 2$, or 1.02, must be subtracted from the heat given for L at 90. The latent heat, therefore, equals $896.4 - 1.02 = 895.38$ B.t.u.

85. Specific Volume of Dry Saturated Steam. — The specific volume, v, of dry saturated steam is the volume occupied by one pound of dry saturated steam. It varies with the pressure, as shown by the curve in Fig. 69. At low pressure the change in specific volume is very rapid. This curve is often called the **saturation curve,** and is useful in determining the condition of a given amount of steam. The volume occupied by a pound of dry saturated steam at any pressure will be twice the volume occupied by half a pound. Neglecting the small volume occupied by water in wet steam, the quality of steam at point b in Fig. 69 is $\frac{ab}{ac}$, or 50 per cent, since only this fraction of the total has been converted into steam.

86. Specific Density of Dry Saturated Steam. — The weight per cubic foot of dry saturated steam is called its specific density. It equals the reciprocal of the specific volume, or $\frac{1}{v}$.

87. Wet Steam, Moisture, and Quality. — When the evaporation of steam takes place rapidly, as is generally the case in a boiler subjected to the intense heat of the furnace, the bubbles of steam formed are large and, upon arriving at the surface of the liquid, burst violently, thus throwing the film of water which surrounds them into the steam space. These particles of water are light and remain suspended in the steam as a mist, or fog. When in this condition steam is called **wet steam,** and a pound of it is composed partly of water and partly of vapor. The term **moisture,** or **priming,** is applied to that portion of the water that remains unevaporated, and the term **quality,** or dryness factor, is applied to that portion of each pound of water actually evaporated into dry steam. Quality and moisture are expressed in per cent of the total weight, and the former is represented by the symbol x. Thus, steam having a quality of 98 per cent is a mixture of ninety-eight parts by weight of steam and two parts by weight of water. The quality of steam can be obtained by subtracting the percentage of moisture from 100 per cent, or vice versa.

88. Total Heat per Pound of Wet Steam. — Equation (24) only applies to dry saturated steam. When the quality is other than unity, the amount of heat added per pound of steam for any given pressure will be less than if the steam were dry, because only a part of the latent heat, L, has been

added. For instance if half a pound of water were evaporated one-half of the latent heat would be added. The whole of the heat of liquid, h, however, is added whether the steam is wet or dry, since the water must be heated to the boiling point before evaporation takes place. The total heat per pound of wet steam, H_w, is therefore the sum of the heat of liquid and the latent heat of evaporation corrected for quality, or, written as an equation,

$$H_w = xL + h \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (25)$$

89. Nature of General Problem. — In general, the problem involved in calculations using steam tables is not one of determining the amount of heat above 32 deg. fahr. in one pound of steam, but in determining how much heat is necessary to form W pounds of steam from water at some temperature other than 32 deg. fahr., or to find how much heat is released by steam when it is condensed.

If the change occurs at constant pressure, and this is the common case, the heat added to the liquid equals the heat of liquid at the absolute pressure under which the liquid is heated, minus the heat of liquid corresponding to the temperature of the water at the start of the addition of heat. This is so because sufficient heat has previously been added to the water to raise it from 32 deg. fahr. to the temperature being considered. In other words, the heat added to the liquid equals $h - h_1$, in which h = heat of liquid at the absolute pressure in boiler, and h_1 = heat of liquid at temperature of water entering boiler.

If the change does not occur at constant pressure, the result can only be obtained by a process of integration.

Example 8. — Find the quantity of heat required to convert 8 lb. of water at a temperature of 182 deg. fahr. and a gage pressure of 150.5 lb. per sq. in. into steam having a quality of 0.98, barometer reading 29.5 inches of mercury.

Solution. — Pressure of the atmosphere $= 29.5 \times 0.491 = 14.5$ lb. per sq. in. abs.

Absolute steam pressure $= 14.5 + 150.5 = 165$ lb. per sq. in.

Total heat per lb. of steam $= (xL + h) - h_1$

From the Steam Tables, page 89, at an absolute pressure of 165 lb. per sq. in.

$L = 858.3$ B.t.u., $h = 337.8$ B.t.u.

h_1 for a temperature of 182 deg. fahr. $= 149.89$ B.t.u., $x = 0.98$

Total heat in 8 lb. of steam $= 8\,[(0.98 \times 858.3 + 337.8) - 149.89] = 8232.3$ B.t.u.

Note. — h_1 can be taken as $(182 - 32)$ without serious error.

90. Superheated Steam. — *Whenever steam exists at a temperature higher than the temperature of saturated steam for that pressure, it is superheated.* The difference in temperature between the temperature of saturated steam, t_v, and the actual temperature of the superheated steam, t_s, is the amount, or **degree of superheat.** For instance, if the absolute pressure of the steam is 200 pounds per square inch and a thermometer placed in it shows a temperature of 510 deg. fahr., the degree of superheat is $510 - 381.9 = 128.1$

deg. fahr., since the temperature of saturated steam at 200 pounds per square inch absolute pressure is 381.9 deg. fahr.

The volume of superheated steam may be obtained with fair accuracy by using the following equation, in which p = absolute pressure in pounds per square inch and v = specific volume:

$$pv^{1.063} = 484.2 \quad \text{(26)}$$

91. Total Heat per Pound of Superheated Steam. — The total heat per pound of superheated steam, H_s, equals the total heat per pound of dry saturated steam plus the heat required to change the temperature from

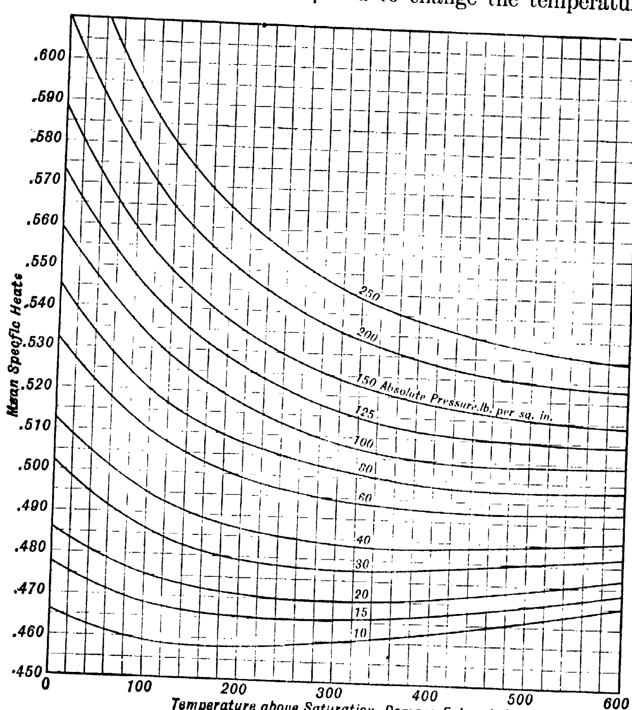


FIG. 70. — Variation of Mean Specific Heat, Superheated Steam.

t_v to t_s. The latter amount of heat equals the specific heat of superheated steam, C_p, times $(t_s - t_v)$.

The total heat per pound of superheated steam can be found from Table 41, page 570, or can be computed from the following equation:

$$H_s = H + C_p\,(t_s - t_v) \quad \text{(27)}$$

The value of the specific heat, C_p, is not constant, but varies with the pressure and temperature. For the range of temperatures encountered in

present practice, a mean value may be used as shown by the curves in Fig. 70. When the pressure is below 100 pounds per square inch, the variation in the mean specific heat is small.

Example 9. — Find the quantity of heat required to produce 10 lb. of steam, superheated 110 deg. fahr., when the pressure is 160 lb. per sq. in. abs. and the temperature of the water from which it is formed is 60 deg. fahr.

Solution. — Using Equation (27) and the Steam Tables, page 89,

$H_s = H + C_p\,(t_s - t_v) = 1167.6 + 0.55\,(110) = 1222.6$ B.t.u.

H at 160 lb. per sq. in. abs. above 60 deg. fahr. $= H - h_{60} = 1195.7 - 28.08 = 1167.6$ B.t.u.; C_p from curve, Fig. 70 $= 0.55$; $t_s - t_v = 110$ deg. fahr.

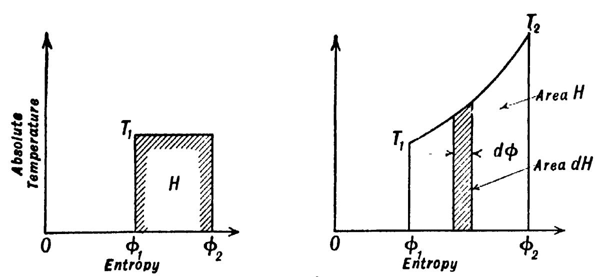
Heat to produce 10 lb. steam $= 1222.6 \times 10 = 12{,}226$ B.t.u.

92. Heat per Pound of Steam, General Case. — Investigators have deduced many mathematical expressions by which this quantity of heat can be obtained. The following equation is most commonly used:

$$H_g = xL + h + C_p \int_{ts}^{tv} dT \quad \ldots\ldots\ldots\ldots \quad (28)$$

When equation (28) is applied to wet or dry steam $C_p \int_{ts}^{tv} dT = 0$, and for superheated and dry and saturated steam $x = 1$.

93. Entropy. — The general heading "Entropy" is given to the last three columns in the saturated steam table. *Entropy may be defined as the change in the quantity of heat per degree of absolute temperature.* During the conversion of water into steam, the heat of liquid and the latent heat of evaporation are given to the pound of water, their sum making the total heat per pound of dry steam. The entropy values to correspond are called entropy of the liquid, θ, entropy of evaporation, L/T, and entropy of steam, ϕ. If the steam is superheated there is a corresponding entropy of superheat, n.



(*a*) *Temperature Constant* (*b*) *Temperature Changing*

FIG. 71. — Entropy Diagrams.

The significance of entropy may be shown by reference to the heat diagram, Fig. 71*a*, the area of which represents heat units. It is bounded by lines representing absolute temperature and entropy.

Entropy Change while Temperature is Constant. — If the temperature is constant at $T°$ absolute, as during the conversion of water into steam, and L heat units are added, the change in entropy, or heat per degree

absolute, is $\phi_2 - \phi_1 = L \div T$. In this case $L = H$ = heat added at the temperature of evaporation.

Example 10. — Calculate the change in entropy during the evaporation of one pound of water at 212 deg. fahr.

Solution. — Since the temperature remains constant during the period of evaporation, the change in entropy equals

$$L \div T = 971.7 \div 671.6 = 1.4469$$

Entropy Change while Temperature Changes. — If the temperature changes while heat is being supplied, as during the addition of the heat of the liquid, the heat area, H, Fig. 71*b*, may be considered as made up of a series of very narrow sections of area dH, for each of which there is a temperature T and a very small width $d\phi$. Then the area $dH = Td\phi$, or $d\phi = dH \div T$; that is, the change in entropy is equal to the small quantity of heat added, divided by the absolute temperature during the addition of the heat. The summation of the elementary heat areas gives the area H. The summation of the $d\phi$ values gives the total entropy change, $\phi_2 - \phi_1$. It can be shown that if the specific heat, C_p, is constant during the addition of the heat, the change in entropy may be written:

$$\phi_2 - \phi_1 = C_p \log_e \frac{T_2}{T_1} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (29)$$

In which T is the initial absolute temperature in degrees Fahrenheit, and T_2 is the final absolute temperature in degrees Fahrenheit.

Example 11. — Calculate the change in entropy of water for the short temperature interval from 65 deg. fahr. to 75 deg. fahr., over which the specific heat is nearly constant and equal to 1. Compare this result with the Steam Table value for the same temperature range and find the entropy of steam at 212 deg. fahr.

Solution. — Using Equation (29)

$$\phi_{75} - \phi_{65} = C_p \log_e \frac{T_2}{T_1} = 1 \log_e \frac{75 + 459.6}{65 + 459.6} = \log_e 1.091 = 0.1899 \text{ or } .019$$

The entropy of water at these temperatures, as given in the Steam Table, is 0.0840 and 0.0650 respectively, and the difference is 0.019, as calculated above.

The *entropy of steam* at 212 deg. fahr. equals the entropy of water plus the entropy of evaporation $= \theta\omega + \frac{L}{T} = 0.312 + 1.4469 = 1.765$.

94. Temperature-entropy Chart. — The temperature-entropy chart is useful for the graphical solution of problems that are tedious and difficult by analytical methods. The lines drawn upon it represent changes in temperature and the corresponding changes in entropy for one pound of water or steam. As previously explained, the area of a closed figure on this chart represents heat units.

In Fig. 72, *vertical distances represent temperature and horizontal distances represent units of entropy.* The temperature scale may be marked t degrees by the thermometer or T degrees absolute. The path *abcd* shows the

changes in temperature and entropy for one pound of water as it is heated in a boiler from 32 deg. fahr. to the temperature corresponding to the pressure maintained, then evaporated at constant pressure into saturated steam at that pressure, and finally superheated by the further addition of heat.

95. Liquid Line. — The line *ab* is known as the **liquid line.** While the water is heated, the entropy increases from ϕ to ϕ_1 and the temperature

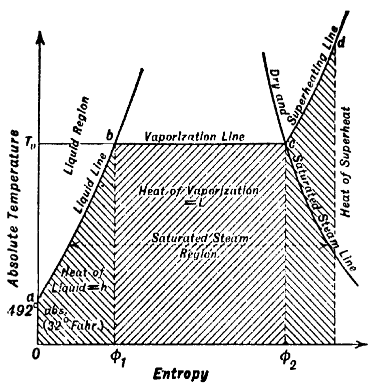


FIG. 72. — Temperature Entropy Diagram.

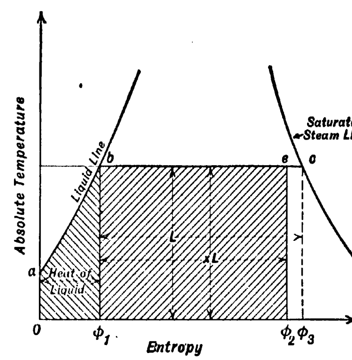


FIG. 73. — Quality from Temperature Entropy Diagram.

from 32 to t_v degrees. The area $O\,ab\,\phi_1$ represents the **heat of the liquid.**

If the vessel containing the water is open to normal atmospheric pressure, 14.7 pounds per square inch absolute, evaporation takes place at 212 deg. fahr. If the vessel is closed and a pressure p, higher or lower than that corresponding to 212 deg. fahr., is maintained in it, the water will receive heat of the liquid until it reaches the temperature corresponding to the pressure. At that point, further addition of heat causes evaporation without an increase in temperature. The liquid line is made up of points representing the entropy and temperature of water at beginning of evaporation.

96. Evaporation, or Latent-heat, Line. — The evaporation line, *bc*, shows the change in entropy while heat is added at constant temperature until evaporation is complete at the point *c*. The area $\phi_1\, bc\, \phi_2 = T_v\,(\phi_2 - \phi_1)$ below the line *bc*, then represents the latent heat *L*, which corresponds to the sum of the internal and the external work done during the formation of steam.

97. Saturation Line. — At the point *c*, the pound of water has been completely evaporated and has taken up all the heat that it can take up at that temperature, and the successive points representing temperature and entropy for complete evaporation make up the saturation or dry-steam **line.**

98. Superheated-steam Lines. — If more heat is added to the pound of dry steam as it passes away from the boiler, its temperature and entropy increase, and this change is shown by the line *cd* sloping up and to the right. *The various lines are plotted by taking the proper value of entropy from the steam table and plotting against the corresponding absolute temperature.*

99. Steam Regions. — The area lying below the liquid line, the evaporation line and the saturation line is called the **saturated-steam region.** Since the area below the liquid line represents the heat of the liquid, h, and the area below the evaporation line represents latent heat, L, the sum of these areas represents total heat H above 32 deg. fahr. If the pound of steam is only partially evaporated, its heat content is shown by the area below the liquid line and below that part of the evaporation line corresponding to the proportion of the pound that is evaporated. Thus, in Fig. 73, $be \div bc$ equals the quality x, and area $oabe\phi_2$ represents $xL + h$.

That part of the temperature-entropy chart lying to the right and above the saturation line is called the **superheated-steam region.** The area below the constant pressure line, *cd*, Fig. 72, corresponds to the heat added to superheat while the temperature and entropy increase.

100. Adiabatic Changes. — During an adiabatic change, there is no change in entropy, since heat is not added or taken away. Lines at right angles to the axis of entropy represent such changes. If a pound of steam having a quality x_1, pressure p, and temperature t expands adiabatically to pressure p_2 and temperature t_2, its quality x_2 at the end of the expansion may be found by equating the values of the entropies for the two conditions and solving for x_2 in the adiabatic equation.

$$\theta_1 + x_1 \frac{L_1}{T_1} = \theta_2 + x_2 \frac{L_2}{T_2} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (30)$$

The value of x_1 being known, the other quantities, except x_2 are taken directly from the Steam Tables.

101. Constant-quality Lines. — Constant-quality lines are made up of points denoting the same quality of steam through the range in temperature and entropy shown on the chart. Thus, if the point 1, Fig. 74, is at 90 per cent of the length of the evaporation line *bc*, and similarly points 2, 3, etc., are at 90 per cent of the lengths of their respective evaporation lines, the constant-quality line *xx*, at 0.90, passes through these points.

Similarly, there may be drawn in the superheated-steam region constant-quality lines yy', or lines joining points at which the number of degrees of superheat is the same through the range of temperature and entropy shown on the chart. *The saturation line and the liquid line are special examples of constant-quality lines.*

102. Constant Total-heat Lines. — Constant total-heat lines are drawn in such a way as to represent the changes occurring in temperature and entropy of a pound of steam while the total heat, or heat content above 32

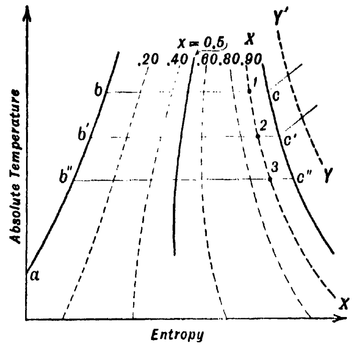


Fig. 74. — Constant Quality Lines.

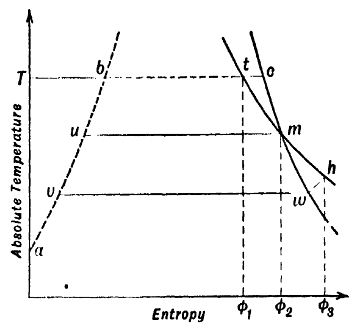


Fig. 75. — Constant Heat Lines.

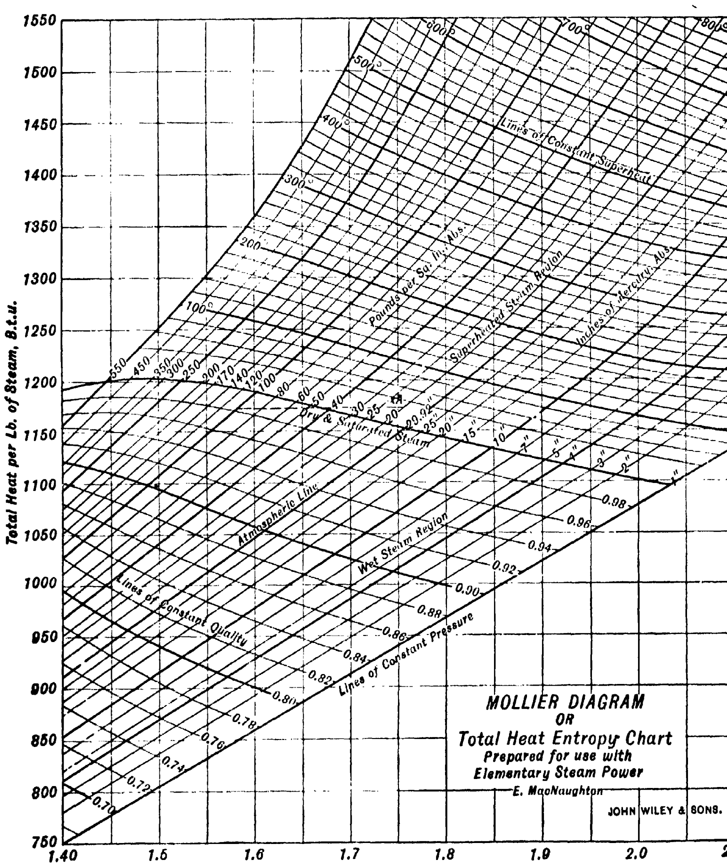


Fig. 76. — Total Heat-Entropy, or Mollier Diagram.

deg. fahr., remains the same. As previously explained, heat units are represented by areas, hence, on the $T-\phi$ chart, Fig. 75, the area below the broken line *abt* represents the heat content of one pound of wet steam H_w. If, for dry steam, the area below *aum* represents the same value of H and for superheated steam the area below *vwh* shows the same total heat, all these areas being the same, the line *tmh* is a line of constant total heat. From the above it is evident that the heat contents which are only sufficient to partially evaporate the steam at the higher temperature and pressure are sufficient to evaporate the steam and produce some superheat at the lower temperature and pressure. Other lines of constant total heat may be drawn in a similar manner. This leads to consideration of the total-heat-entropy diagram, one use for which will be explained.

103. Total Heat Entropy Diagram. — *By using values of total heat as vertical distances, and values of entropy as horizontal distances, a graphical chart can be prepared on which constant-pressure lines and constant-quality lines are drawn, for use in the graphical solution of problems.*

A small section of such a chart is shown in Fig. 76. Horizontal lines are lines of constant total heat, and vertical lines are lines of constant entropy. The saturation line is shown as a heavy line separating the region of saturated steam from that of superheated steam. Lines which slope downward and to the right are constant-quality or constant-superheat lines, and lines which slope upward, and to the right are lines of constant pressure.

Example 12. — If steam at an absolute pressure of 100 pounds per square inch is expanded without loss of heat to 20 pounds absolute, and the number of degrees of superheat at the lower pressure is $t_s - t_v = 20$, the condition of the steam is shown at xA. To find its quality before expansion, follow the constant-total-heat line until it intersects the pressure line at 100 pounds per square inch absolute. The quality corresponding is thus found to be 0.982.

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PROBLEMS AND REVIEW QUESTIONS

1. For the complete burning of one pound of carbon, 2.66 lb. of oxygen are required. What weight of air is necessary to supply this amount of oxygen?

2. Find the quantity of heat in B.t.u. required to heat 10 lb. of air from 60 deg. fahr. to 550 deg. fahr. the pressure being constant.

3. Find the weight of 100,000 cu. ft. of air at a temperature of 125 deg. fahr. and at 14.7 lb. per sq. in. abs.

CHAPTER V

STEAM CALORIMETERS

104. Foreword. — The amount of moisture in a pound of steam is ordinarily determined by a calorimeter. There are several types of calorimeters, and they are classified, according to their accuracy, as: throttling, combined throttling and separating, separating, electric, and barrel. The **throttling** and **separating calorimeters** will be described, as they are the types commonly used in test work.

105. Throttling Calorimeter. — The throttling calorimeter, recommended by the American Society of Mechanical Engineers is shown in Fig. 77. It is made of pipe fittings and consists essentially of two tees in which thermometer wells are placed. A disk, held between two flanges and having a $\frac{1}{8}$-inch orifice, is located between the tees. The calorimeter is attached to the steam line by a perforated pipe known as a **sampling nozzle,** with a globe valve placed between the calorimeter and the steam line to shut off the calorimeter when desired. The discharge from the calorimeter is directly into the atmosphere. A steam gage, for obtaining the pressure of the steam, is attached by a water siphon to the steam line near the calorimeter.

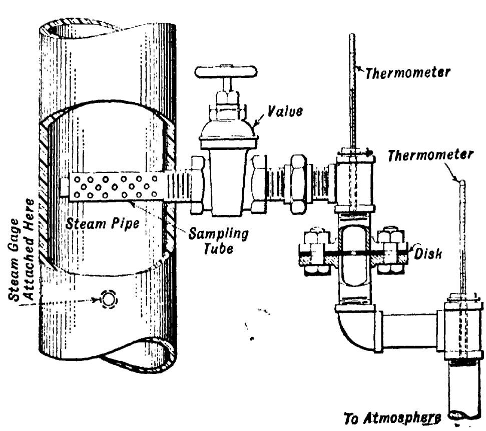
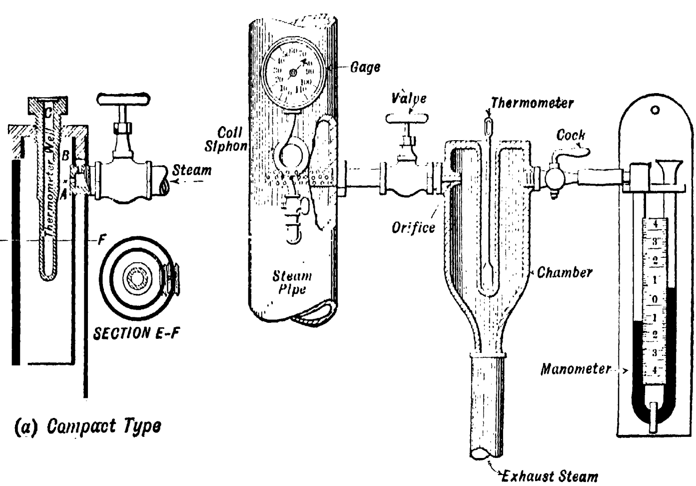


Fig. 77. — A. S. M. E. Type Throttling Calorimeter.

Operation. — Steam from the steam pipe passes through the sampling nozzle around the first thermometer cup. It then passes through the $\frac{1}{8}$-inch orifice and around the second thermometer to the atmosphere. The thermometer wells should be filled with a heavy cylinder oil. The instrument should be thoroughly warmed; and the temperature of the steam, after passing the orifice as indicated by the second thermometer, should become stationary before any moisture determinations are made.

Principle of Operation. — This instrument works upon the principle that steam, in passing through an orifice, such as that of the calorimeter, does so without performing external work. Hence, the quantity of heat in a pound of steam, under the conditions existing in the calorimeter, will be the same as that in a pound of wet steam at the absolute pressure in the steam main, provided there is no radiation loss. Expressing the above differently, *a pound of wet steam at a high pressure contains more heat than a pound of wet steam at a lower pressure, and as external work is not performed by the steam, when passing through the orifice, the surplus heat evaporates the moisture and superheats the steam.* For instance, take the higher pressure as 150 lb. per sq. in. abs. and the lower pressure as 14.7 lb. per sq. in. abs. The total heat per pound of dry steam at the former pressure is 1194.7 B.t.u., and at the latter, 1151.7 B.t.u., a difference of 43 B.t.u. The number of degrees of superheat would therefore be 43 B.t.u. divided by the specific heat of superheated steam (0.47) which equals 91.5 deg. fahr. Moisture in the steam at the higher pressure would reduce the amount of the superheat shown above.

106. Other Forms of Throttling Calorimeters. — For most testing work, the upper thermometer, Fig. 77, is dispensed with, and the pressure in



(b) Common Type

Fig. 78. — Other Types of Throttling Calorimeters.

the steam main is measured by an accurate pressure gage. The calorimeter then resembles one of the forms shown in Fig. 78.

The calorimeter shown in Fig. 78*a* has two concentric metal cylinders screwed to a cap containing a thermometer well. Steam is supplied to

the annular space between the two cylinders, to act as a jacket and reduce, as far as possible, the radiation losses from the inner cylinder. The inner chamber is open at the bottom, and hence the pressure inside the calorimeter is considered as atmospheric.

The form shown in Fig. 78*b* is quite common. It consists of a chamber so shaped that it provides a thermometer well in the center and has an orifice at one side through which steam enters the calorimeter chamber. The pressure inside the calorimeter is measured by a mercury manometer. A gage cock is placed between the calorimeter and the manometer, to permit cutting the manometer out of service. The discharge from the chamber is from the bottom, through suitable piping, to the atmosphere.

The calorimeter and the pipe connections to the steam line should be thoroughly insulated with hair felt, to reduce the losses caused by radiation to a minimum. This precaution is necessary with any type of calorimeter.

107. Method of Computing Moisture and Quality. — The method explained is for the form of calorimeter shown in Fig. 78*b*. The following relation holds true, as has been shown under Art. 106: *total heat per pound of wet steam before passing the orifice = total heat per pound of superheated steam after passing the orifice.*

Let H_w = total heat per pound of steam at absolute steam-pipe pressure.

L = latent heat per pound of steam at absolute steam-pipe pressure.

h = heat of liquid per pound of steam at absolute steam-pipe pressure.

H = total heat per pound of dry saturated steam at absolute pressure, after passing the orifice.

t_v = temperature of saturated steam at the absolute pressure in the calorimeter.

t_s = temperature of superheated steam in the calorimeter.

0.47 = specific heat of superheated steam at atmospheric pressure.

x = proportion by weight of dry steam per pound of steam.

m = proportion by weight of moisture = $1 - x$.

The total heat per pound of wet steam before passing the orifice = $xL + h$, and the total heat per pound of superheated steam after passing the orifice = $H + C_p(t_s - t_v)$; consequently

$$xL + h = H + C_p(t_s - t_v), \text{ and } x = \frac{H + C_p(t_s - t_v) - h}{L} \quad (31)$$

or, when written in the following form, Equation (31) may be used to find the proportion of moisture directly:

$$m = 1 - x = \frac{H_w - H - 0.47\,(t_s - t_v)}{L} \quad \ldots . \quad (32)$$

Quality may also be found, without using the above equations, from the total-heat-entropy diagram, as explained under Art. 103, page 101.

Example 13. — The pressure in a steam line to which a calorimeter is attached is 150 lb. per sq. in. gage; pressure in calorimeter atmospheric; temperature in calorimeter, 262 deg. fahr.; barometer, 30.02 in. mercury. Find the quality of the steam.

Solution. — Pressure of the atmosphere = 30.02 × 0.491 = 14.74 lb. per sq. in.
Absolute steam pressure in line = 150 + 14.74 = 164.74 lb. per sq. in.
Temperature corresponding to calorimeter pressure (t_v) = 212.15 deg. fahr.
Temperature in calorimeter (t_s) = 262.0 deg. fahr.
Specific heat of superheated steam (C_p) at atmospheric pressure = 0.47.
Latent heat (L) corresponding to absolute steam-pipe pressure = 858.4 B.t.u.
Heat of liquid (h) corresponding to absolute steam-pipe pressure = 337.7 B.t.u.
Total heat per pound of dry steam (H) at absolute calorimeter pressure = 1151.8 B.t.u.
Using Equation (31) with proper substitutions

$$x = \frac{H + C_p\,(t_s - t_v) - h}{L} = \frac{1151.8 + 0.47\,(262 - 212.2) - 337.7}{858.4} = 0.975$$

Moisture = m = $(1 - x)$ 100 = 2.50 per cent.

108. Sources of Error, Throttling Calorimeter. — There is a slight error due to the value taken for the specific heat of superheated steam at atmospheric pressure; this error, however, is negligible. The thermometer stem is ordinarily not heated throughout its full length. Moreover, the thermometer may have an initial error, and there may be radiation losses from it.

The amount of heat available for superheating the steam in the calorimeter is reduced by radiation, and hence the instrument will not give the true quality where appreciable radiation occurs. In order to correct for the thermometer and radiation error, the Power Test Committee of the American Society of Mechanical Engineers recommend referring the readings, as found during the determination, to a **"normal" reading** of the low-pressure thermometer corresponding to dry steam. The *"normal" reading* should be determined by attaching the calorimeter to a horizontal steam pipe, in such a way that the sampling nozzle projects upward nearly to the top of the pipe, there being no perforations in the nozzle and the steam entering only through its open upper end. The test should be made with the steam in a quiescent state and with the steam pressure maintained as observed in the main trial, the calorimeter thermometer to be the one used on the trial or one exactly similar.

By means of this "normal" reading, the radiation correction is calculated as follows:

Let T denote the normal reading for the conditions existing in the trial. *The effect of radiation from the instrument will be to lower the temperature of the steam at the calorimeter pressure.* Let m_1 represent the proportion of water in the steam which will lower its temperature an amount equal to the loss by radiation.

Then $$m_1 = \frac{H_w - H - 0.47\,(T - t_v)}{L} \qquad (33)$$

This amount of moisture, m_1, was not in the steam originally, but is the result of condensation in the instrument, through radiation. Hence, the true amount of moisture in the steam, M, is the difference between the amount as determined in the trial and that resulting from condensation, or,

$$M = m - m_1 = \frac{H_w - H - 0.47\,(t_s - t_v)}{L} - \frac{H_w - H - 0.47\,(T - t_v)}{L}$$

$$= \frac{0.47\,(T - t_s)}{L} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (34)$$

The theoretical readings for dry steam, where there are no losses due to radiation, are obtainable from Equation (32) by putting $m = 0$ and solving for t_s. The difference between the theoretical reading for dry steam and the normal reading for no moisture will be the thermometer and radiation correction to be applied in order that the correct reading of t_s may be obtained.

For any calorimeter within the range of its ordinary use, such a thermometer and radiation correction taken from one normal reading is approximately correct for any conditions, with the same or a duplicate thermometer.

The percentage of moisture in the steam, corrected for thermometer error and radiation, would be determined as follows: Assume the absolute pressure to be 195 pounds per square inch and the thermometer reading to be 295 deg. fahr. A normal reading, taken in the manner described, gives a value of $T = 303$ deg. fahr. The percentage of moisture, corrected for radiation, is

$$m = \frac{0.47\,(T - t_s)}{L} = \frac{0.47\,(303 - 295)}{846.0} = 0.0045, \text{ or } 0.45 \text{ per cent.}$$

Limits. — The limits of moisture within which the throttling calorimeter will work are, at sea level, from 2.88 per cent moisture at 50 pounds gage pressure, to 7.17 per cent moisture at 250 pounds pressure.

109. Separating Calorimeter. — The separating calorimeter, Fig. 79, consists of a cast-iron body so constructed that there is an inner and an outer vessel with a space between. This space forms a steam jacket for the inner cylinder and serves to prevent radiation from the inner vessel, except that which takes place from the gage-class connections. The inner chamber has no direct connection with the outer jacket space, but is connected at one side to a water glass having an attached scale graduated to read in hundredths of a pound of water. This scale is corrected for expansion, to give correct readings with steam at 100 pounds per square inch gage pressure.

At the top of the inner chamber is a small cup having projecting fins which assist in separating the moisture from the steam. The bottom of the cup is closed, but there is an opening into the inner chamber under

each fin. The outer chamber is connected to a steam gage which indicates the pressure in the outer vessel. This gage is graduated, by trial, to read the weight of steam passing through the instrument in ten minutes. At the bottom of the outer chamber is a small orifice of known size, through which the dry steam passes from the calorimeter.

A cap, having a deflecting plate and a pipe which extends well down into the cup, or separator, is attached to the top of the body by a threaded connection. An angle valve connects this cap with the sampling nozzle.

Principle of Operation. — *The separating calorimeter mechanically separates the moisture from the steam and collects it in a chamber where its amount may be accurately determined.* It depends for its accuracy upon the complete separation of the entrained water from the steam, and upon the accuracy of the gage calibrations.

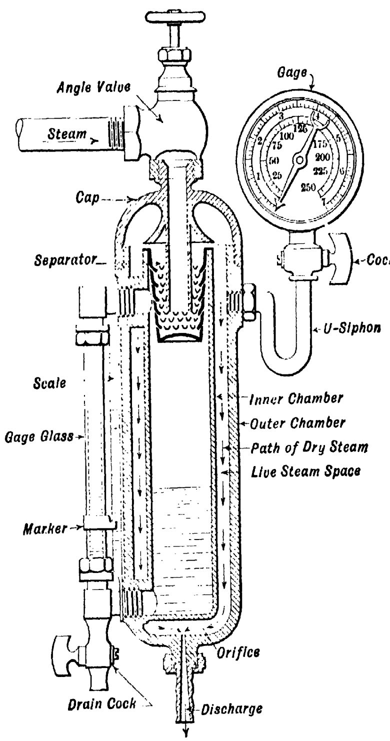


Fig. 79. — Separating Calorimeter.

Operation. — When the angle valve is opened, steam from the sampling nozzle passes downward into the perforated cup, where its direction is reversed. The moisture, being heavier than the steam, is left in the cup and falls into the inner chamber where its amount may be read on the graduated scale, or it may be drained into a suitable vessel and weighed. The dry steam, passing upward, enters the outer chamber through a small opening between the inner vessel and the cap. It then flows through the small orifice at the bottom to the exhaust pipe. The amount of steam flowing through the orifice can be read on the gage dial, which should be calibrated whenever used; or its weight may be computed by **Napier's Law**, which is as follows: *the flow of steam from a higher to a lowre pressure is proportional to the higher absolute steam pressure, as long as the lower pressure is less than* 0.58 *of the higher pressure.* Expressed as an equation

$$W = \frac{Pa}{70} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (35)$$

in which

W = weight of steam, pounds per second.

P = absolute steam pressure before passing the orifice, pounds per square inch.

a = area of the orifice, square inches.

This equation will give accurate results, provided the orifice is not clogged with sediment and its area has been accurately obtained. The amount of steam passing through the instrument may be determined more accurately by attaching a hose to the steam outlet, the hose extending into a vessel containing water and resting on a scale. The steam flowing will then be condensed by the water and its amount determined by weight. The weight of moisture in pounds can be read on the graduated scale, or weighed as desired.

The proportion of moisture can be calculated from the following relation:

$$\text{Moisture} = \frac{\text{weight of moisture}}{\text{weight of dry steam plus weight of moisture}} = \frac{w}{W + w} \qquad (36)$$

The quality of the steam is found by subtracting the proportion of moisture from 1.

Example 14. — During the test of a 9 × 16 × 24 × 24 in. steam engine, the separating calorimeter attached to the exhaust pipe of the high-pressure cylinder gave the following data: weight of moisture in ten minutes, 0.13 lb.; weight of dry steam in ten min., 1.775 lb.; pressure in calorimeter, 29.5 lb. per sq. in. gage; barometer, 29.45 in. mercury. Find: (*a*) quality of steam and (*b*) weight of steam passing orifice in 10 min., using Napier's equation.

Solution. — (*a*) Using Equation (36) with proper substitutions:

$$\text{The proportion of moisture} = \frac{w}{W + w} = \frac{0.130}{1.775 + 0.13} = \frac{0.130}{1.905} = 0.068, \text{ or } 6.8 \text{ per cent.}$$

$$\text{Per cent quality} = (1 - 0.068)\ 100 = 93.8$$

(*b*) To find weight of steam passing the orifice use Equation (35)

$$W = \frac{Pa}{70} = \frac{44 \times 0.0047}{70} = 0.00295 \text{ lb. per second.}$$

$P = 29.5 + 14.5 = 44$ lb. per sq. in. abs.

$a = 0.0047$ sq. in. (by measurement).

Weight of steam in ten minutes $= 0.00295 \times 60 \times 10 = 1.770$ lb.

Limits. — Theoretically, this instrument is not limited to any specific range. If it is desired to have a check upon its accuracy, a throttling calorimeter may be attached to the steam outlet; the total moisture will then equal the sum of that given by the two instruments.

110. Sampling Nozzle. — The principle source of error in steam calorimeter determinations is the failure to obtain an average sample of the

steam delivered by the boiler, and it is doubtful whether such a sample is ever obtained. The two governing factors in obtaining such a sample are the type of sampling nozzle used and its location.

The American Society of Mechanical Engineers recommends a sampling nozzle made of ½-inch iron pipe closed at the inner end, the interior portion perforated with not less than twenty ⅛-inch holes equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than ½ inch from the inner wall of the pipe. Many engineers object to the use of a perforated sampling nozzle because it ordinarily indicates a higher percentage of moisture than is actually present in the steam. This is due to the fact that if the perforations come close to the inner surface of the pipe, the moisture, which in many instances clings to this surface, will flow into the calorimeter and cause a large error. Where a perforated nozzle is used, generally, it may be said that the perforations should be at least one inch from the inner pipe surface.

A sampling nozzle, open at the inner end and unperforated, undoubtedly gives as accurate a measure as can be obtained of the moisture in the steam passing that end. It would appear that a satisfactory method of obtaining an average sample of the steam would result from the use of an open-end unperforated nozzle passing through a stuffing box which would allow the end to be placed at any point across the diameter of the steam pipe.

111. Location of Sampling Nozzle. — The calorimeter should be located as near as possible to the point from which the steam is taken, and the sampling nozzle should be placed in a section of the main pipe near the boiler, turbine, or engine under test, and at a point where there is no chance of moisture pocketing in the pipe. The American Society of Mechanical Engineers recommends that a sampling nozzle should be located in a vertical main, rising from the boiler with its closed end extending nearly across the pipe. Where non-return valves are used, or where there are horizontal connections leading from the boiler to a vertical outlet, water may collect at the lower end of the uptake pipe and be blown upward in a spray which will not be carried away by the steam, owing to a lack of velocity. A sample taken from the lower part of this pipe will show a greater amount of moisture than a true sample. With gooseneck connections, a small amount of water may collect on the bottom of the pipe near the upper end, where the inclination is such that the tendency to flow backward is ordinarily counterbalanced by the flow of steam forward over its surface; but when the velocity momentarily decreases the water flows back to the lower end of the goose-neck and increases the moisture at that point, making it an undesirable location for the sampling nozzle. In any case, it should be borne in mind that with low velocities, there is a tendency for drops of entrained water to settle to the

bottom of the pipe, and to be temporarily broken up into spray whenever an abrupt bend or other disturbance is met.

REFERENCES

Power Plant Testing, MOYER.
Steam, BABCOCK AND WILCOX Co.
Mechanical Laboratory Methods of Testing, SMALLWOOD
Power Test Code, A. S. M. E.

PROBLEMS AND REVIEW QUESTIONS

1. The pressure in a steam main is 150 lb. per sq. in. gage. Barometer reads 29 in. of mercury. Steam is allowed to expand through an orifice to a pressure of 25 lb. per sq. in. gage. Temperature of steam after expanding is 293 deg. fahr. What was the quality of steam in the main pipe? Check, using Mollier chart.

2. A steam gage attached to a steam pipe reads 110 lb. per sq. in. What is the maximum amount of moisture in the steam which a throttling calorimeter can detect, if 10 degrees superheat is required and the discharge is directly into the atmosphere, where the barometer reads 29.2 in. of mercury?

3. The temperature in a throttling calorimeter, attached to a steam main, is 275 deg. fahr. The pressure in the steam main is 100 lb. per sq. in. gage. Barometer, 29.5 in. of mercury. Pressure in the calorimeter is atmospheric. Find the quality of steam in the steam main.

4. Derive the equation used in calculating the quality of steam when using a throttling calorimeter. Explain the meaning of each symbol.

5. State the principle of operation of (*a*) the throttling calorimeter, (*b*) the separating calorimeter.

6. The following data apply to a separating calorimeter attached to a steam main: Diameter of orifice, 0.04 in.; pressure in calorimeter, 116 lb. per sq. in. absolute; moisture collected in 10 minutes, 0.2 lb.; barometer, 29.48 inches mercury. Find the quality of the steam.

7. A separating calorimeter is attached to a steam main in which the pressure is 120 lb. per sq. in. absolute. The moisture collected in 15 minutes is 1.17 lb. Find the quality if the area of the orifice is 0.011 sq. in. and the barometer, 29.9 in. mercury.

CHAPTER VI

FUELS

112. Foreword. — Without a knowledge of fuels and their characteristics it is impossible to purchase them intelligently or use them economically. Such knowledge is especially necessary when one is purchasing a fuel to be used with existing equipment or when choosing equipment for burning different grades of fuels.

The cost of coal, which is the most common fuel for steam power plants, varies between 40 and 70 per cent of the total operating expense of a plant. As the price of coal advances, a more careful selection must be made, and, to assist in making such a choice, information should be at hand regarding ash and moisture content, size and storage requirements of the available fuels, and particularly regarding the working qualities of the fuel under the conditions where it is to be burned.

113. Definition of a Fuel. — A fuel is any material that is combustible; that is, any material that can be burned, and that may be obtained in quantities at a reasonable price. It may be natural or artificial.

114. Classification of Fuels. — Fuels used for power purposes may be classified as:

1. Solid — Coal; Wood; Vegetable wastes
2. Liquid — Crude petroleum and its distillates
3. Gaseous — Natural; Artificial: Blast-furnace gas, Coke-oven gas, Producer gas, Illuminating gas

The solid fuels, especially coal, are used most extensively. Wood is generally too expensive to be used for fuel purposes. Bagasse, a sugar-cane refuse, is used on sugar plantations in the South. Crude oil is used in sections adjacent to oil fields and also on shipboard. The demand for this fuel is increasing rapidly, because of the increasing cost of coal. Natural gas and blast-furnace gas are used to a limited extent.

115. Formation of Coal. — Coal is of vegetable origin, and is the remains of prehistoric forests. Destructive distillation, together with great pressure *has resolved the organic matter into its ultimate constituents, carbon, hydrogen, oxygen and other substances,* in varying proportions. The factors that have

produced the different grades of coal are, (1) time, (2) depth of bed, (3) disturbance of bed, and (4) intrusion of mineral matter.

116. Classification of Coals. — Coals may be classified according to their origin, as has been done by the United States Geological Survey, Table 8.

TABLE 8.—GEOLOGICAL CLASSIFICATION OF COAL

Graphite.........	Not commonly used as fuel.
Anthracite.......	Domestic use principally.
Semi-anthracite...	Most common for power purposes.
Semi-bituminous..	
Bituminous......	
Sub-bituminous ..	
Lignite..........	
Peat	

117. Composition of Coals. — Coal consists of carbon, hydrogen, moisture, ash, oxygen, nitrogen, and sometimes sulphur. The carbon, hydrogen and sulphur are the " combustible " in the coal, a term applied in boiler practice to that part of the fuel which is left after taking away the moisture and ash, thus including whatever nitrogen and oxygen may be present. The remaining substances are impurities which lower the value of a coal as a fuel.

118. Coal Analysis. — Two methods of analysis are used to determine the composition of coal. They are known as the **proximate analysis,** which determines the composition of coal by mechanical processes, and the **ultimate analysis,** which separates coal into its chemical constituents by chemical porcesses. The analysis to be used depends in a large measure upon the use which is to be made of the data obtained.

119. Proximate Analysis. — In the proximate analysis of coal, as received, five determinations are usually made; namely, moisture, volatile matter, ash, fixed carbon, and sulphur. These constituents are expressed in per cent by weight; the sum of the first four items is taken as 100 per cent, with the sulphur separately determined.

Moisture. — This determination does not differentiate between the moisture that comes from external sources and that which is inherent in the coal. The amount of moisture in such coals as anthracite and semi-bituminous may be obtained by placing a 15 or 20 pound sample in a shallow pan and drying, at about 90 deg. fahr., for twelve hours. A more accurate method is to heat a small quantity of finely ground coal, in a double-walled air bath, to a temperature between 240 and 280 deg. fahr. until the sample reaches a minimum weight. For lignites, a lower drying temperature is used. *The loss in weight is called moisture.*

Volatile Matter. — *The volatile matter consists of the carbon combined with*

the hydrogen, together with the other gas-forming constituents which are driven off by heat.

One gram of finely powdered coal from which the moisture has been removed is placed in a platinum crucible with a tight cover. The crucible and contents are heated to about 1750 deg. fahr., for seven minutes. The crucible is then cooled and weighed. *The loss in weight is the volatile matter.*

Ash. — *Ash is the incombustible residue from the complete combustion of the coal.* The sample used in the volatile matter determination is heated in an open pan, by a blast lamp, until completely burned. A stream of oxygen may be used to hasten combustion. *The weight of the residue is ash.*

Fixed carbon. — *Fixed carbon is the carbon which is not in combination with any other element.* It does not represent all the carbon in the coal, as a considerable amount is driven off with the volatile matter in combination with hydrogen and oxygen. It is not pure carbon, as it contains some ash-forming constituents, and approximately one-half the sulphur. *The weight of fixed carbon is obtained by subtracting the weight of moisture, ash, and volatile matter from the weight of the original sample.*

Sulphur. — This element may be separately determined by burning a portion of the sample with a suitable chemical mixture that will combine with the sulphur in such a form that the sulphur can be separated into a sulphur compound and its amount determined by weighing.

120. Ultimate Analysis. — This analysis expresses the composition of a coal in percentage by weight of carbon, hydrogen, nitrogen, sulphur, oxygen, and ash. It requires the careful use of chemical apparatus, and reliable results can be obtained only by those entirely familiar with all details of the work. *The sum of these five constituents is generally taken as equal to* 100 *per cent,* but moisture is sometimes included.

121. Method of Reporting Analyses. — Both analyses may be expressed in terms of

(1) Coal " as received " or " **as fired.**"

(2) Coal " moisture free " or " **dry.**"

(3) Coal " moisture-and-ash-free " or " **combustible.**"

The first term under each method is that used by the BUREAU OF MINES, and the second that used in the A. S. M. E. TESTING CODE. The latter will be used in this text.

Engineers prefer to have the analyses expressed for coal " as fired," since it represents the condition of the coal as fed to the furnace. The dry-coal basis is however, generally used for reporting analyses, as it excludes the moisture, which is principally due to weather conditions during shipment. A better idea as to the value of the coal can be obtained with the moisture excluded.

Coal " as fired " is in the same condition as when it comes from the

bunkers. The analysis on a dry-coal basis is found from the analysis "as fired," by dividing each constituent by (1 − proportional weight of moisture). The analysis on a combustible basis is found by dividing the analysis "as fired" by [1 − (proportional weight of ash + moisture)].

Example 15. — The per cent by weight of the constituents of a Kentucky coal "as fired" is given by the following ultimate analysis: ash 4.39, sulphur 1.22, hydrogen 5.43, carbon 77.37, nitrogen 1.83, oxygen 9.76. Moisture from proximate analysis 3.10. Convert this analysis to the dry-coal and the combustible basis.

Solution. — The ultimate analysis of a coal "as fired" when reported as above, contains the free moisture as a part of the hydrogen and oxygen. To obtain the moisture as a separate item, one-ninth of the moisture should be subtracted from the hydrogen and eight-ninths from the oxygen. The reason for this is that water is composed of one part hydrogen and eight parts oxygen. The first column in the solution has been corrected by this method.

Dividing the analysis "as fired" by (1 − the proportional weight of moisture) to obtain the analysis on the dry basis, and by [1 − (the proportional weight of ash + moisture)] for the combustible basis, there results:

	Coal as fired	Coal dry	Combustible
Carbon.................	77.37	79.84	83.63
Hydrogen..............	5.07	5.24	5.48
Nitrogen..............	1.83	1.89	1.98
Oxygen................	7.02	7.24	7.59
Sulphur................	1.22	1.26	1.32
Ash.....................	4.39	4.53	
Moisture...............	3.10		
	100.00	100.00	100.00

122. Use of Analyses. — A knowledge of the constituents of a coal, as given by the ultimate analysis, indicates its adaptability for a given use. It also furnishes data from which the heat value may be calculated and permits classification on the basis of (1) **fixed carbon,** (2) **the ratio of carbon to hydrogen,** and several other similar classifications. It should be borne in mind that the analyses do not furnish all the information necessary to determine the value of a coal for a particular plant, as will be explained later.

123. Heat Value of Solid Fuels. — *The heat value of a fuel is the amount of heat a pound of the fuel will generate when completely burned.* It may be determined (1) by **calorimeter,** (2) by **computation based on ultimate analysis,** and (3) from **curves based on the proximate analysis.**

124. Description of Fuel Calorimeter. — A standard type of fuel calorimeter, known as the Mahler Bomb Calorimeter, is shown in Fig. 80. It may be used for solid or liquid fuels. The calorimeter has a porcelain-lined steel bomb having a tight-fitting cover screwed into place. Within the bomb is a small platinum pan for holding the fuel. Two electrodes, con-

nected at their lower end by a fuse wire, extend through the cover of the bomb, and connect the fuse wire to an electric circuit containing a suitable switch and a few dry batteries. The bomb sets into a calorimeter which contains a definite weighed amount of water. A stirring device for agitating the water within the calorimeter is attached to a support connected to the outside of the apparatus. The temperature of the water in the calorimeter is shown by an accurate thermometer graduated to read to one-thou-

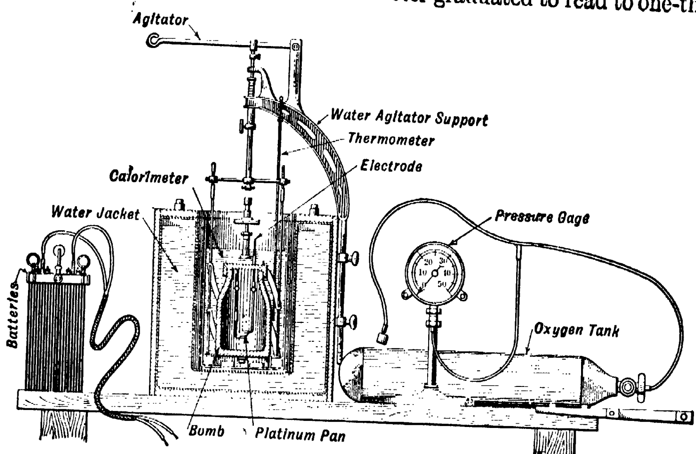


FIG. 80. — Bomb Fuel Calorimeter.

sandth of a degree. The calorimeter and bomb are surrounded by an outer vessel covered with asbestos and filled with water at a temperature slightly higher than the surrounding atmosphere. This outer vessel maintains constant conditions and makes the determination of the radiation loss more accurate. An oxygen tank supplies the oxygen required to burn the fuel, and a pressure gage attached to the oxygen tank piping is used to determine the pressure of the oxygen entering the bomb.

Operation. — From an average sample of pulverized coal, approximately one gram is placed in the fuel pan, and a piece of fine iron fuse wire, which dips into the coal, is connected between the electrodes. The cover of the bomb is then screwed tightly into place and the bomb is placed in the calorimeter, which has been partially filled with a definite weight of water — generally an amount of water equal to the **water equivalent** of the apparatus, as determined by calculation or experiment. Oxygen from the tank is then slowly admitted to the bomb until the pressure is about 25 atmospheres.

Temperature readings of the water in the calorimeter are taken at one minute intervals, for a period of time long enough to insure a constant rate of change. The coal is then ignited by closing the electric circuit. After

ignition, temperature readings are taken at the instant of closing the electric circuit, at one-half minute intervals for five minutes, and at one-minute intervals thereafter for a sufficient length of time to determine the rate of change after combustion is complete. From the temperature readings taken before ignition and after the five-minute interval, the radiation correction is calculated.

The temperature range during the period of burning is corrected for radiation, and is then multiplied by the sum of the water equivalent of the calorimeter and bomb and the weight of the water in the calorimeter, to give the heat of combustion expressed in gram-calories, weight being in grams and temperature in degrees Centigrade. From this product are subtracted the heat resulting from the formation of aqueous nitric acid, the heat produced by the combustion of the sulphur to sulphuric acid, and the heat formed by the combustion of the fuse wire.* The remainder, divided by the weight of the sample in grams and multiplied by 1.8, gives the heat in B.t.u. per pound.

125. High and Low Heat Value of Fuel. — With fuels which contain hydrogen, the heat value as found by the calorimeter is higher than that realized under working conditions in boiler practice, by an amount equal to the latent heat of evaporation of the water which is formed. This heat reappears when the vapor is condensed, but in ordinary conditions of combustion, the vapor passes away uncondensed.

The **"higher" heat value** is that determined by the calorimeter, and should be used in all boiler test calculations. The **"lower" heat value** is the higher heat value less a correction for the heat lost in the escaping steam.

126. Heat Value from the Ultimate Analysis. — The equation most used for calculation of heat values from the composition of coal is that proposed by Dulong, and is written:

$$\text{Heat per pound of fuel, B.t.u.} = 14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000S\dagger \qquad (37)$$

in which C, H, O, S, = proportion by weight of the carbon, hydrogen, oxygen and sulphur, as shown by the ultimate analysis of a pound of fuel, and the coefficients = the heat evolved when one pound of carbon, hydrogen, and sulphur are completely burned.

Part of the hydrogen in the coal is assumed to be in combination with the oxygen in the coal. The hydrogen available for producing heat is therefore the actual hydrogen minus that already combined with the oxygen

* These corrections are small and are often neglected.

† Bulletins of the Bureau of Mines give Dulong's formula as:

$$\text{B.t.u. per lb.} = 14{,}544\,C + 62{,}028\left(H - \frac{O}{8}\right) + 4050S.$$

to form water. This latter equals one-eighth of the oxygen present, or $\frac{O}{8}$. The **available,** or **free hydrogen,** as it is called, is written $(H - \frac{O}{8})$.

Example 16. — The ultimate analysis of a coal, expressed in per cent by weight, gave: carbon, 84.45; hydrogen, 4.25; oxygen, 3.04; nitrogen, 1.28; sulphur, 0.91; and ash, 6.07. Find the heat per pound of coal by Dulong's equation.

Solution. — Using Equation (37)

$$\text{Heat per pound of coal} = 14{,}600\ C + 62{,}000\ (H - \frac{O}{8}) + 4000\ S.$$

Substituting the value of carbon, hydrogen, oxygen and sulphur, there results

$$\text{Heat per pound of coal} = 14{,}600 \times 0.845 + 62{,}000\ (0.0425 - \frac{0.0304}{8}) + 4000 \times 0.0091$$
$$= 14{,}772 \text{ B.t.u.}$$

127. Heat Value from Proximate Analysis. — Calculations of the heat value of a coal based on a proximate analysis are not satisfactory. A curve showing the relation between the heat value per pound of combustible of a coal and the percentage of fixed carbon is given in Fig. 81. It is based on data taken from tests conducted by the United States Geological Survey.

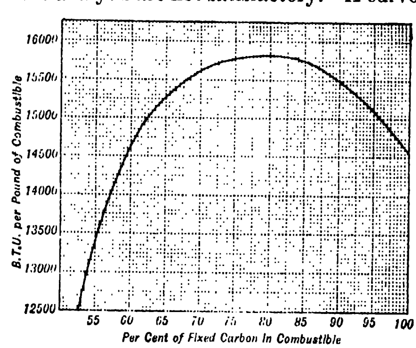


FIG. 81. — Heat Value of Coal using Proximate Analysis.

128. Methods of Expressing Heat Value of Coal. — The heat value of a coal may be expressed in terms of (1) *coal as fired,* (2) *dry coal,* or (3) *combustible.*

The heat value per pound of dry coal is obtained from the heat value per pound of coal as fired, by dividing by (1 − *proportional weight of moisture*). It represents the heat that would be generated by burning one pound of coal containing no moisture. The heat value per pound of combustible is found by dividing the heat value per pound of coal as fired, by [1 − (*proportional weight of ash* + *moisture*)]. It represents the heat that would be generated by burning one pound of coal containing neither ash nor moisture.

Example 17. — The heat value per pound of a North Dakota coal *as fired* is 7,069 B.t.u., with moisture 35.96 per cent and ash 7.75 per cent. Find the heat value per pound of (*a*) dry coal, (*b*) combustible.

Solution. — Dividing the heat value as fired, by (1 − *proportional weight of moisture*) for the heat value of dry coal, and by [1 − (proportional weight of ash + moisture)] for heat value of combustible.

$$\text{Heat per pound of dry coal} = \frac{7{,}069}{1 - 0.3596} = 11{,}031 \text{ B.t.u.}$$

$$\text{Heat per pound of combustible} = \frac{7{,}069}{1 - (0.3596 + 0.0775)} = 12{,}560 \text{ B.t.u.}$$

129. Classifications of Coals Based on Analysis. — Coals are frequently classified by the **carbon content** and by the **ratio of carbon to hydrogen,** Table 9.

TABLE 9. — CLASSIFICATION OF COALS BY CARBON CONTENT AND CARBON-HYDROGEN RATIO*

Name of Coal	Percentages of Combustible		Carbon — Hydrogen Ratio
	Fixed Carbon	Volatile Matter	
Anthracite	97 to 92.5	3 to 7.5	30 to 26
Semi-anthracite	92.5 to 87.5	7.5 to 12.5	26 to 23
Semi-bituminous	87.5 to 75	12.5 to 25	23 to 20
Bituminous, East	75 to 60	25 to 40	20 to 14.4
Bituminous, West	65 to 50	35 to 50	14.4 to 11.2
Lignite	50 and under	50 and over	11.2 to 9.3
Peat			9.3 to....

* Or ratio of the total carbon to the hydrogen.

130. Characteristics of Coals. — Typical analyses of coals, together with the fusing point for the lower grades of coal, are given in Table 10.

TABLE 10. — ANALYSES OF TYPICAL COALS

	Name of Coal						
	Anthracite	Semi-Anthracite	Semi-Bituminous	Eastern Bituminous	Western Bituminous	Sub-Bituminous	Lignite
Proximate Analysis							
Moisture	3.45	1.57	4.07	8.98	14.43	22.63	36.78
Volatile matter	2.75	9.40	16.34	34.49	29.48	35.68	28.16
Fixed carbon	87.90	83.69	68.47	50.30	42.81	37.19	29.97
Ash	5.90	5.34	11.12	6.33	13.28	4.50	5.09
Ultimate Analysis:							
Carbon	88.86	85.46	76.51	70.50	54.59	54.91	41.87
Hydrogen	2.04	3.72	4.27	4.76	5.49	6.39	6.93
Nitrogen	0.90	1.12	1.00	1.36	1.11	1.02	0.69
Oxygen	1.95	3.45	6.59	15.66	21.52	32.59	44.94
Sulphur	0.35	0.91	0.51	1.39	4.01	0.59	0.48
Ash	5.90	5.34	11.12	6.33	13.28	4.50	5.09
Heat Value:							
Calorimeter	13,950	....	13,509	12,417	10,064	9,734	7,002
Dulong's equation	14,103	14,552	13,329	12,084	9,866	9,478	6,944
Classification:							
Carbon-hydrogen ratio	42.50	23.0	19.60	14.80	12.30	9.40	9.60
Fusing point of ash*	....	....	2600°F.	2300°F.	2100°F.	2200°F.	....

* Approximate valves

Anthracite Coal has a deep black color, and high luster. It is a hard coal, composed almost entirely of fixed carbon, with a specific gravity of 1.3 to 1.9. It burns with a short, bluish flame, and does not swell when burning. The commercial sizes of anthracite are given in Table 11, and are graded by the size of wire-mesh screen through which the sample will pass or will not pass. A $\frac{1}{4}$-inch mesh screen has openings $\frac{1}{16}$ of a square inch in area.

TABLE 11. — SIZES OF ANTHRACITE, OR "HARD," COAL

Name of Size	Size of Screen through which Coal	
	Will Pass	Will Not Pass
Culm	$\frac{3}{32}$-in.	......
Birdseye	$\frac{5}{16}$-in.	$\frac{1}{4}$-in.
Buckwheat No. 1	$\frac{1}{2}$-in.	$\frac{1}{4}$-in.
Buckwheat No. 2, or Rice	$\frac{1}{4}$-in.	$\frac{1}{8}$-in.
Pea	$\frac{3}{4}$-in.	$\frac{1}{2}$-in.
Chestnut	$1\frac{3}{8}$-in.	$\frac{3}{4}$-in.
Stove or Range	2-in.	$1\frac{3}{8}$-in.
Egg (in the East)	$2\frac{3}{8}$-in.	2-in.
Broken	4-in.	$2\frac{1}{2}$-in.

Semi-anthracite coal has an iron-black color, is not as hard as anthracite and has less luster. It is a free-burning coal, and burns with a longer and more luminous flame than anthracite. While not very plentiful, it is the best steam producing coal found in the United States. Its specific gravity is about 1.4.

Semi-bituminous coal is a softer coal than anthracite, with a lower specific gravity and more volatile matter. It kindles readily and gives off a small amount of smoke when burning. This coal is extensively used in steam power plants.

Bituminous coal is soft, having a color ranging from pitch black to dark brown, with a silky or resinous luster. It has a large amount of volatile matter, is brittle, breaks into small pieces when stored, and burns with a long yellow and smoky flame. Its specific gravity is about 1.3.

Bituminous coals may be classified into **coking** or **caking,** and **non-coking** coals. The latter burn freely, do not fuse and are known as free-burning coals. Coking coals swell up, become pasty and fuse together when burning. They are rich in hydrocarbons and hence are valuable for the manufacture of gas. These coals absorb moisture readily. They are sized as in Table 12.

Cannel coal is homogeneous, has a grayish-black color with a resinous luster, is high in hydrocarbons, kindles readily, and burns with a dense, smoky flame. It is used for the manufacture of gas and in fireplaces. Its specific gravity is about 1.24, and its heating value is low.

TABLE 12. — SIZES OF BITUMINOUS, OR "SOFT," COAL

Name of Size	Size of Screen through which Coal	
	Will Pass	Will Not Pass
Run of the Mine	Mixture of lumps and fine coal, or slack.	
Sized Coal,* such as:		
Lump	6-in.	3-in.
Egg	3-in.	1½-in.
Nut	½-in.	⅜-in.
Slack	⅜-in.	......
Screenings	Smallest sizes	

* The sizes given differ in different parts of the United States.

Sub-bituminous coal is a grade between true bituminous coal and lignite. It resembles bituminous coal in appearance, is not woody like lignite, and slacks and absorbs moisture readily.

Lignites have a brown color with a pitchy luster and woody structure. Lignite is high in ash, moisture and oxygen, with a specific gravity between 1.2 and 1.23, and cannot be transported far because it breaks so easily. It burns freely with a bright, slightly smoky, yellow flame, and is used principally for the manufacture of gas.

Peat is an accumulation of partly decomposed water plants, mosses, and other vegetable matter. It has a color ranging from yellow to reddish brown, a fibrous texture and a high moisture content, and must be dried before burning.

131. Powdered, or Pulverized, Coal. — Pulverized coal is coal that has been reduced to a fineness such that 95 per cent will pass through a 100-mesh screen and 85 per cent through a 200-mesh screen. When in this condition it is an impalpable powder that will float in air. The term "pulverized" is ordinarily used to mean coal that is too finely divided to be used in any other way. The satisfactory use of coal in this form, however, depends upon the fineness to which it has been pulverized.

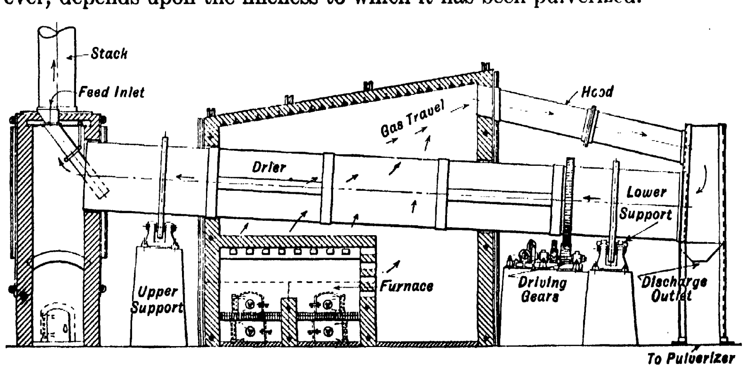


FIG. 82. — Pulverized Coal Drier.

The coal is first crushed and passed through an inclined dryer, Fig. 82, which consists of a steel shell rotated by suitable gearing; the moisture is thus removed before the coal is pulverized. The shell is located in a furnace and so arranged that the hot gases from the furnace circulate around the outside of the dryer shell. They then pass through a hood to the lower end of the shell and return through the shell, in direct contact with the coal, to the chimney at the upper end of the shell. Coal is fed into the upper end and travels slowly to the lower end, where it is discharged to the top of the pulverizer, Fig. 83, and is fed into the pulverizing zone by a screw conveyor. The coal passes to the grinding chamber where it is reduced to a fine powder by the centrifugal force produced by four hardened steel balls which are caused to rotate on a hardened steel race. This fine powder is picked up by a fan, located above the grinding element, and is discharged through screens, near the top, into a narrow passage connected at the bottom to a chamber in which a fan is located. This fan discharges the pulverized coal to a suitable conveyor, which carries it to an air-tight container from which the coal is fed to the boiler. The cost of pulverizing varies from 20 to 60 cents per ton, including drying and conveying.

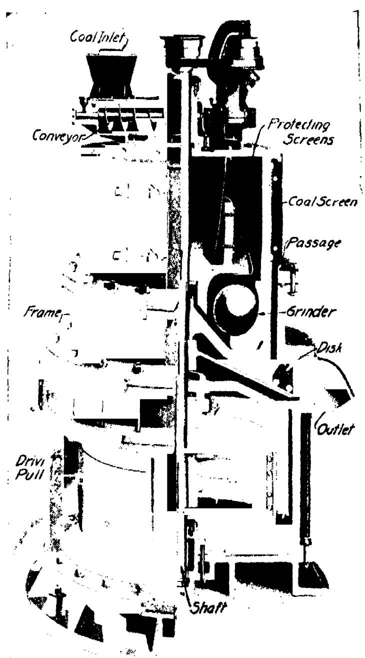


FIG. 83. — Fuller Mill for Pulverizing Coal.

The **advantages** claimed for coal in this form are:

1. More complete burning of the fuel because of better mixture with the air for combustion.
2. No limit to the grade of coal.
3. Reduction in labor of handling coal and ashes.
4. Elimination of standby losses.
5. Adaptability to varying loads.
6. Easy control of air supply.

The **disadvantages** are:

1. *Difficulty of reducing coal to uniform size.*
2. Impossibility of introducing coal into furnace uniformly.
3. Danger of fuel clogging feed pipes, if moisture is not removed.
4. Liability to explosion (which is small when the coal is properly handled).
5. Decrease in efficiency, if tubes are not kept free from ash.
6. Slagging of ashes and deterioration of furnace, if temperature is not properly controlled.
7. Caking of coal in storage.
8. Impracticability of applying to plants burning less than 100 tons per day.

132. Briquetted Coal. — This fuel consists of finely ground coal, mixed with a suitable binder, such as pitch, and pressed into briquettes. Such treatment decreases the fuel loss through the grates, increases the heat value of a fuel and permits the use of lower-grade fuels. Although the cost and difficulty of manufacture have prevented the extensive use of this fuel as a source of power, briquetted lignites are used extensively in some regions for domestic purposes. Briquetting permits the weathering, handling and burning of lignites without disintegration.

133. High- and Low-ash Coals. — Coals having **over ten per cent of ash** are classed as high-ash coals, and those having **less than ten per cent** as low-ash coals. Coals having a high-ash content require more labor for the removal of ash from the furnace.

134. Clinkering and Non-clinkering Coals. — Clinker is formed by the mechanical combination of the constituents which form the ash, or by the fusing of the ash; it may be hard or soft. Hard clinker, which is produced by the melting of the ash, forms in hard lumps; soft clinker, which is formed by the silica of the ash combining with the fusible constituents, iron and lime, in the ash, remains molten and continues to grow. Ash, having a melting temperature of 2700 deg. fahr., will not form clinker at ordinary furnace temperatures. *The clinkering qualities of a coal may be of more importance than the heat value.*

Clinker causes furnace linings and grate bars to wear out rapidly. It reduces furnace efficiency and capacity, besides entailing considerable work in its removal. *Hard clinker may be formed by poor firing methods.*

Clinker may be prevented by (1) carrying a thin fire, (2) avoiding the improper use of slice bar and rake in mixing ash into the burning fuel, (3) firing in small amounts on thin spots of fire, (4) having ashpit clean and doors open, (5) using water in the ashpit or steam jets under grates to prevent clinker sticking to grates, and (6) avoiding shaking coal that will burn in ashpit.

135. Effect of Ash, Moisture, Sulphur, and Oxygen in Coals. — **Ash** is composed of silica, alumina oxide, iron oxide and other impurities in small amounts. It reduces the heat value of a coal by replacing combustible matter, and adds to the cost of handling and storage. Additional expenses resulting from the presence of ash are the cost of cleaning fires and ashpit and the expense of disposal of the ash. Ash obstructs the flow of air through the grates and may form troublesome clinker under certain conditions.

Moisture lowers the heat value of a coal, because heat is required for its evaporation; it reduces the actual weight of coal purchased by the weight of moisture, and adds to the cost of transportation. Small sizes of coal hold much more moisture than large sizes.

Sulphur occurs in coal as iron pyrite, marcasite, sulphate of lime, iron and alumina, as organic sulphur, and sometimes as free sulphur uncombined with other elements. *As free sulphur it has its full heat value. When it occurs as a sulphate, it has no heat value*, and in any case the available hydrogen must be high for sulphur to have any appreciable heat value. In small amounts it may assist in the prevention of clinker; but if the ratio of the weight of sulphur in the ash to the total ash is high, it may assist in forming a troublesome clinker.

Oxygen has about the same effect upon the heat value of a coal as an equivalent amount of ash. It is an original impurity of which the better grades of coal have small amounts.

136. Storage and Weathering of Coal. — The storage of coal has become a necessity, because of market conditions, danger of labor difficulties, and the crowding of transportation facilities. Anthracite is almost an ideal coal for storing; it is not subject to spontaneous ignition and may be stored in large piles. Bituminous coal is likely to ignite if placed in deep piles and will also suffer from disintegration. It is sometimes stored under water to prevent **spontaneous ignition,** that is, ignition resulting from the heat produced by the absorption of oxygen by the hydrocarbons in the coal and by the slow combustion, or oxidation, of carbon, sulphur, and available hydrogen. Spontaneous ignition can only take place when the air supply is sufficient to support oxidation but will not entirely remove the heat produced.

Coal loses heat value when exposed to air during storage. When coal is stored in piles, its temperature should be closely watched; if it is becoming too high the pile should be moved. Powdered coal can be stored for only a short time, and never in the open air.

137. Location of Coal Fields. — Anthracite coals are nearly all found in northern Pennsylvania, with some distribution in small areas in some of the Western States. Semi-anthracite is only found in small quantities in southern Pennsylvania, West Virginia, and Virginia. The centers of pro-

duction of semi-bituminous coals are the Pocahontas and New River fields in Virginia and West Virginia, Georges Creek field in Maryland, Windber field in Pennsylvania, and small areas in Arkansas, Washington, and Colorado. Bituminous coal fields are found in the Appalachian mountains and scattered through the Middle Western and Western states. Sub-bituminous fields are located mostly in North Dakota. Lignites are found in North Dakota and most of the Western states, and in Alaska.

138. Purchase of Coal. — Coal is ordinarily purchased by weight; but when weight is the only consideration the results are likely to be unsatisfactory, because the heat value of the coal and its working qualities are of prime importance in the operation of the plant. A more satisfactory method of purchase is by specification, the contract to cover coal of a definite analysis and heat content, with suitable allowance in price for variation therefrom.

139. Liquid Fuel. — Petroleum is the only liquid fuel sufficiently abundant and cheap to be used for the generation of steam. It is generally classified according to the base it yields upon distillation; that is, (1) **paraffin,** (2) **asphalt,** and (3) **olefine.**

To the first group belong the fuels of the Appalachian Range and the Middle West of the United States. This group yields so many valuable light oils that the price is prohibitive for use as a fuel.

The asphalt group is found in Texas and California. They have a color varying from reddish brown to jet black, and are extensively used for fuel. The olefine group is found in Russia and is used as a fuel oil.

Crude oils consist of carbon and hydrogen, with varying amounts of moisture, sulphur, nitrogen, arsenic, phosphorus and silt. The moisture content varies from 1 to 30 per cent and affects the heat value of the fuel. Analyses of several typical fuel oils are given in Table 13.

TABLE 13. — COMPOSITION AND HEAT VALUE OF TYPICAL FUEL OILS

Location of Oil Well	Per Cent Carbon	Per Cent Hydrogen	Per Cent Sulphur	Per Cent Oxygen	Specific Gravity	Flash Point ° F.	B.t.u. per Lb.
California..........	81.52	11.51	0.55	6.92*	....	230	18,667
Texas, Beaumont...	84.60	10.90	1.63	2.87	0.924	180	19,060
Pennsylvania.......	84.90	13.70	....	1.40	0.886	...	19,210
Russia, Caucasus...	86.60	12.30	...	1.10	0.938	...	20,138

* Includes nitrogen.

140. Comparison of Coal with Oil. — The **advantages of oil** over coal, when used as a fuel under boilers, are: (1) reduction in cost of handling, (2) saving of labor throughout the plant, (3) reduction in storage space required, (4) higher efficiencies and capacities, (5) easy regulation of load,

(6) no loss in heat value when properly stored, (7) cleanliness and freedom from dust.

Disadvantages of oil are: (1) danger of explosions, (2) possibility of larger upkeep, when furnaces are not properly designed and operated.

141. Gaseous Fuels. — These fuels are used in certain localities for steam-generating purposes and in manufacturing enterprises. As previously mentioned, the principal gaseous fuels are: (1) natural gas, and (2) artificial gas, such as blast-furnace, coke-oven, and producer gas.

Natural gas is a product of nature and, in the coal district, consists chiefly of methane and ethylene with some hydrogen and nitrogen. A typical analysis of gas from the Pittsburgh region, in per cent by volume, gave the following results: hydrogen 20.02, methane 72.18, carbon monoxide 1.00, carbon dioxide 0.80, oxygen 1.10, heavy hydrocarbon 4.30. In the South Western oil fields, large quantities are available from gas wells. This "**casing-head**" gas, as it is called, from which gasoline is sometimes made, consists of about 79 per cent methane and 18 per cent ethylene. *The higher heating value of natural gas varies from* 950 *to* 1000 *B.t.u. per cubic foot.*

Blast-furnace gas is the waste product from furnaces used to smelt iron ores. For each ton of iron produced, about 10,000 pounds of gas is obtained. A typical analysis of a blast-furnace gas, expressed in per cent by volume, is as follows: carbon dioxide 9.85, nitrogen 53.92, oxygen 0.36, carbon monoxide 32.73, hydrogen 3.14. *Its heat value is due primarily to carbon monoxide and varies from* 85 *to* 100 *B.t.u. per cubic foot* of gas.

By-product coke-oven gas is a product of destructive distillation of coal in a distilling coke oven. The gases from the oven, instead of being burned at the point of origin, as in the bee-hive coke oven, are removed from the oven through an uptake pipe, and cooled, yielding tar, ammonia, illuminating gas and fuel gas, as by products. This gas resembles natural gas more closely than blast-furnace gas does. A typical analysis, expressed in per cent by volume, is as follows: carbon dioxide 3.20, oxygen 0.4, carbon monoxide 6.3, methane 29.60, hydrogen 41.6, nitrogen 16.1. *The heat value varies from* 400 *to* 500 *B.t.u. per cubic foot.*

Illuminating gas is generally a carburetted water gas, which is made by the decomposition of steam into hydrogen and carbon monoxide by contact with incandescent carbon. Hydrocarbon gases from oil or naphtha are then mixed with it to give the illuminants—particles of carbon which become incandescent.

Producer gas is a mechanical mixture of carbon monoxide, hydrogen, nitrogen, and small amounts of methane, ethylene, oxygen and carbon dioxide. It is made from the incomplete combustion of anthracite coal, bituminous coal, coke or peat, and also from briquettes of anthracite slack and lignite.

142. Heat Value of Gaseous Fuels. — The heat value of gaseous fuels is generally expressed in B.t.u. per cubic foot, and may be obtained by:

1. Calculation, based upon the chemical analysis of the gas.
2. Experiment, made with a gas calorimeter.

Heat Value Based on Analysis. — The most satisfactory method of determining the heat value of a gas is to use the percentages of the constituent gases and the heat value of each constituent gas, as determined by experiment, and as given in Table 14.

TABLE 14. — VOLUME AND CALORIFIC VALUE OF VARIOUS GASES

Name of Gas	Symbol	Volume Cu. Ft. per Lb.	B.t.u. per Cu. Ft.	Volume of Air per Cu. Ft. of Gas
Hydrogen	H	177.90	349	2.41
Carbon Monoxide	CO	12.81	347	2.39
Methane	CH_4	22.37	1053	9.57
Ethylene	C_2H_4	12.80	1675	14.33
Ethane	C_2H_6	11.94	1862	16.74
Acetylene	C_2H_2	13.79	1556	11.93

Example 18. — The analysis of a natural gas taken from an oil well in Anderson, Indiana, gave percentages by volume as follows. Find the heat value of the oil.

Hydrogen, H	1.86	Nitrogen, N	3.02
Methane, CH_4	93.07	Oxygen, O	0.42
Carbon monoxide, CO	0.73	Heavy hydrocarbons	0.47
Carbon dioxide, CO_2	0.26	Hydrogen sulphide, H_2S	0.15

Solution. — Using the heat values of the various combustible constituents as given in Table 14, and multiplying by the weight of each constituent in a cubic foot of the gas, the following results are obtained:

From hydrogen	$0.0186 \times 349 =$	6.50
From methane	$0.9307 \times 1053 =$	980.02
From heavy hydrocarbons	$0.0047 \times 1364 =$	6.41
From carbon monoxide	$0.0073 \times 347 =$	2.53
B.t.u. per cu. ft.		995.46

143. Junker Gas Calorimeter. — This instrument is shown in Fig. 84, and consists of a vertical cylindrical water chamber containing vertical tubes, heated by the gas burned in a Bunsen burner placed underneath the water chamber. The products of combustion pass upward through a combustion chamber and downward through the tubes, while water passes continuously in at the bottom and out at the top. The quantity of gas is measured by a gas meter, and the water that passes is collected and weighed. The temperature of the entering and leaving water is measured by thermometers. The heat of combustion per cubic foot of gas is determined by multiplying the weight of water in pounds by the rise of temperature in

degrees Fahrenheit and dividing the product by the volume of the gas used in cubic feet. This result is corrected for the moisture in the gas and reduced to equivalent heat value at 60 deg. fahr. and 30 inches barometer,

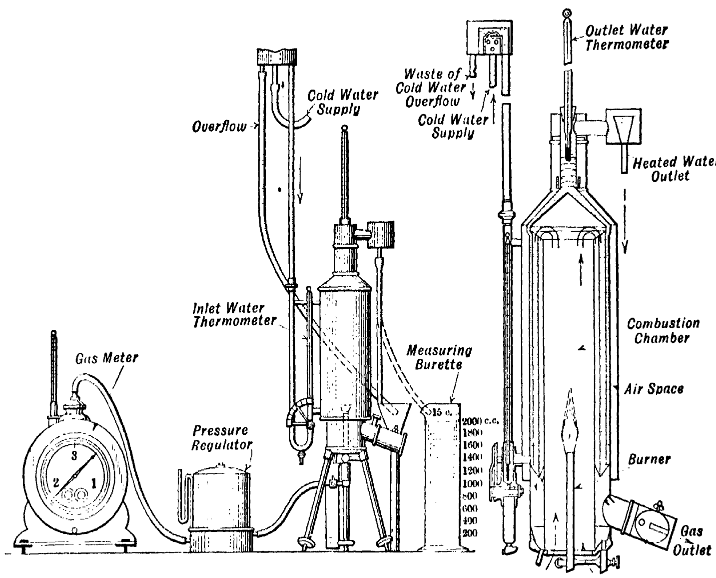


FIG. 84. Junker Gas Calorimeter, Assembly and Sectional View.

and is the "higher" heat value. The "higher" heat value is used in all test work, unless otherwise mentioned.

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PROBLEMS AND REVIEW QUESTIONS

1. What information is obtained from a proximate analysis of coal? Explain what each item represents.

2. How is a proximate analysis made? Explain fully.

3. What constituents of a coal are determined by the ultimate analysis?

4. Give three methods used to report an analysis.

5. The analysis of a coal from the Connellsville mine in Pennsylvania gives the following, based on coal "as fired":

Proximate analysis: moisture 5.13, volatile matter 27.87, fixed carbon 58.29, ash 8.71.

Ultimate analysis: sulphur 0.86, hydrogen 4.91, carbon 73.13, nitrogen 1.50, oxygen 10.89.

Convert the above analyses to (*a*) dry, (*b*) combustible basis.

6. The proximate analysis of anthracite coal gives: moisture, 5.41; volatile matter, 7.02; fixed carbon, 71.79; ash, 15.78; sulphur, 0.74.

Convert to a dry basis.

7. What is the heat value of a fuel? How is it determined?

8. The ultimate analysis of an Illinois coal "as fired" is: sulphur, 4.06; hydrogen, 4.95; carbon, 53.40; nitrogen, 0.89; oxygen, 16.61.

From the proximate analysis the moisture was 10.69 and the ash 20.09.

Calculate by Dulong's equation the heat value of (*a*) dry coal, (*b*) coal as fired.

9. The ultimate analysis of a Kentucky coal "as fired" is: sulphur, 1.22; hydrogen, 5.43; carbon, 77.37; nitrogen, 1.83; oxygen, 9.76; moisture, 3.10; ash, 4.39.

Calculate by Dulong's equation the heat value of (*a*) dry coal, (*b*) coal as fired.

10. The proximate analysis of New River West Virginia coal "as fired" is: moisture, 3.34; volatile matter, 21.25; fixed carbon, 73.18; ash, 2.23. Find from Fig. 81, page 119, the heat value of the coal per pound of combustible.

11. Coals from various parts of the United States have heat values "as fired" and moisture contents as shown below. Calculate the heat values on: (*a*) dry, (*b*) combustible basis.

Locality	Moisture	Ash	Heat value "as fired" B.t.u.
Colorado	18.68	5.99	10,143
Illinois	8.31	10.49	11,727
Maryland	3.42	7.09	14,162
Missouri	8.33	19.36	10,586
Montana	4.13	30.86	9,095

12. Give the physical characteristics of four common coals.

13. How is anthracite coal sized? Bituminous coal?

14. What is powdered coal and what advantages has it? What disadvantages?

15. Explain the cause of clinker.

16. What is the effect of the following in a coal: (*a*) ash, (*b*) moisture, (*c*) sulphur, (*d*) oxygen.

CHAPTER VII

COMBUSTION, FLUE GAS ANALYSIS, BOILER LOSSES

144. Foreword. — Combustion, or burning, is a process in which a substance unites with oxygen, with the evolution of heat and often of light. At the instant of burning, the fuel must be in the gaseous form, to unite with the oxygen, which is generally supplied by the air. A rapid combustion, such as that which takes place in the cylinder of a gas engine, is called an explosion. An illustration of imperfect combustion is that of a smoldering fire in a peat bog.

The burning, or oxidation, of a gaseous fuel is accomplished by mixing air or oxygen with it in the proper proportions, and at the right temperature. To burn a liquid fuel, suitable means must be provided to vaporize or atomize it. The process of gasifying or atomizing a solid fuel and mixing it with air at the proper temperature, is attended with more difficulty, involving also a method of disposal of ash and waste. When in the powdered form, a solid fuel must be burned while suspended in the necessary air for combustion.

Progress in the study of the combustion of fuels has been made largely through the development of instruments for indicating furnace conditions. By application of these instruments and attention to their indications, improved conditions of combustion have followed in many power plants, with a resulting increase in economy in the use of fuel.

145. Requirements for Perfect Combustion. — There are three conditions necessary for perfect combustion;

1. An ample supply of air.
2. A thorough mixing of the air and gases.
3. A sufficiently high temperature to maintain combustion.

The successful solution of the problem of combustion is dependent upon the fulfilment of these requirements.

146. Stages of Combustion of Solid Fuel. — In the combustion of a solid fuel, three stages are usually recognized, namely:

1. Heating the fuel to the temperature required for ignition, or kindling point, Table 15.
2. Expelling the volatile gases, which are then burned.
3. Burning the solid remainder to ash.

These may be briefly stated as **absorption, distillation** and **oxidation.**

TABLE 15. — TEMPERATURES AT WHICH VARIOUS COMBUSTIBLES IGNITE *

Name of Combustible	Temp. °F.	Name of Combustible	Temp. °F.
Phosphorus	150	Fixed carbon, bituminous coal	766
Sulphur	470	Fixed carbon, semi-bituminous	870
Carbon monoxide, CO	1210	Fixed carbon, anthracite	925
Hydrogen	1130	Cannel coal	688
Methane, marsh gas, CH_4	1202	Dried peat	435
Ethylene, olefiant gas, C_2H_4	1022	Lignite dust	300
Ethane, C_2H_6	1000		
Acetylene, C_2H_2	900		

* From STROMEYER'S Marine Boiler Construction, and COSGROVE'S Coal.

The combustion of fuel on a grate may be explained by reference to Fig. 85. Consider the fuel to be coal. When fresh coal is thrown on an incandescent fuel bed, the moisture in the coal is driven off as steam. The heat then distils off the hydrocarbon gases, which unite with oxygen from the air admitted above the grate, and, under favorable conditions, are completely

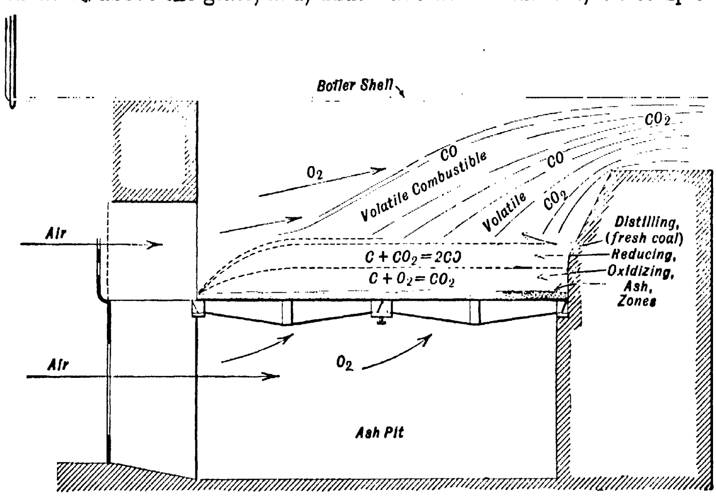

FIG. 85. — Illustrating Changes occurring during Combustion of a Solid Fuel.

burned. The fuel which remains consists of fixed carbon and ash. The carbon in the lower layers, where sufficient air is present below the grate, is burned to **carbon dioxide,** CO_2. As the carbon dioxide passes through the middle layer, where there is excess of carbon, it loses oxygen and becomes **carbon monoxide,** CO. *This is called reduction, or loss of oxygen.* Then the carbon monoxide, passing away over the fuel bed, should receive oxygen again from air admitted over the fire and be completely burned to

carbon dioxide. *The zones in the fuel bed are the distillation, reduction, oxidation and ash zones.*

147. Combustible Constituents of Coal. — The combustible constituents of coal may be grouped as **volatile combustible** and **fixed carbon.**

Volatile Combustible. — The light hydrocarbon gases, called **methane,** or **marsh gas,** CH_4, and **ethylene,** or **olefiant gas,** C_2H_4, existing in coal at ordinary temperatures, are first to pass off when the fuel is heated. The tarry hydrocarbon substance remaining is then further decomposed, principally into marsh gas, olefiant gas, and carbon in the form of soot. Part of the sulphur in chemical combination is also volatilized and burns.

Fixed Carbon. — The coke remaining after the volatile combustible has passed off consists of fixed carbon and ash. The fixed carbon is the principal combustible of coal, and the steaming value of bituminous and semi-bituminous coals increases, up to a certain point, with the percentage of fixed carbon they contain.

148. Incombustible Constituents of Coal. — *The incombustible constituents of the coal are the ash, consisting principally of silicon, aluminum, calcium, iron, and magnesium, together with small amounts of oxygen and nitrogen.* The oxygen is regarded as being in chemical combination with one-eighth its weight of hydrogen, the remainder of the hydrogen being available for the production of heat. The nitrogen, in passing away with the gases, instead of adding heat value, takes away the heat required to raise its temperature to that of the waste gases. Part of the sulphur remains in combination in the ash.

149. Fundamentals of the Chemistry of Combustion. — A knowledge of the chemical changes resulting from the union of oxygen with carbon, hydrogen and sulphur is essential to an adequate presentation of the subject of combustion.

All substances are composed of atoms, which are minute particles of chemical elements, arranged in groups called molecules. The elements are represented, for convenience, by symbols; thus, the symbols, C, H, N, O, S, represent one atom each of carbon, hydrogen, nitrogen, oxygen and sulphur. These elements, together with iron (Fe), silicon (Si), calcium (Ca), aluminum (Al), and magnesium (Mg), are the principal constituents of coal.

Molecules of the simple gases, such as hydrogen, oxygen, and nitrogen, have 2 atoms each and *are represented by the symbols,* H_2, O_2, *and* N_2, respectively. The atoms of carbon, sulphur, and iron are found in combination with other elements and have no separate molecular symbol given to them.

When substances unite chemically to form a new combination, they unite in definite proportions by weight, new molecules being formed by a re-distribution of the atoms.

The molecular weights of the substances formed are the sum of the atomic weights. The atomic weights are purely relative and were arbitrarily established. Hydrogen, as the lightest known substance, was given the atomic weight of 1 and the weights of the other elements were fixed in relation to it. Carbon, which is 12 times as heavy as hydrogen, was given an atomic weight of 12, and oxygen, being 16 times as heavy, was given the atomic weight of 16. Later determinations have shown that oxygen is less than 16 times as heavy as hydrogen, and as a result, hydrogen has been given an exact atomic weight of 1.008. Oxygen, with an atomic weight of exactly 16, is now used as the basis of all atomic weights.

The atomic and molecular weights of substances entering into combustion are given in Table 16.

TABLE 16. — ATOMIC AND MOLECULAR WEIGHTS OF SUBSTANCES ENTERING INTO COMBUSTION

Substance	Atomic Symbol	Atomic Weight	Molecular Symbol	Molecular Weight
Hydrogen	H	1	H_2	2
Carbon	C	12	—	—
Oxygen	O	16	O_2	$2 \times 16 = 32$
Nitrogen	N	14	N_2	$2 \times 14 = 28$
Sulphur	S	32	—	—
Iron	Fe	56	—	—
Carbon monoxide	—	—	CO	$12 + 16 = 28$
Carbon dioxide	—	—	CO_2	$12 + 32 = 44$
Methane, or marsh gas	—	—	CH_4	$12 + 4 = 16$
Ethylene, or olefiant gas	—	—	C_2H_4	$24 + 4 = 28$
Gaseous water, or steam	—	—	H_2O	$2 + 16 = 18$

150. Chemical Reactions. — The chemical union of substances may be expressed by an equation, or reaction, as it is called. Thus, if two molecules of hydrogen unite with one molecule of oxygen, forming water vapor, the equation may be written in the **atomic form** as:

$$2H + O = H_2O \qquad (38)$$

; or using the **molecular form,** for reasons to be explained, and writing the molecular weights below each molecule,

$$\begin{aligned} &2H_2 + O_2 = 2\,H_2O \qquad (39)\\ &4 \quad + 32 = 36, \text{ or } 1 + 8 = 9 \end{aligned}$$

From Equation (39) the following information regarding the weight and volume of the substances entering into the reaction may be obtained:

1. Weight. — It is evident from these figures that 4 parts by weight of hydrogen unite with 32 parts by weight of oxygen, or 1 pound of hydrogen unites with 8 pounds of oxygen to form 9 pounds of water vapor.

2. Volume. — By **Avogadro's Law,** the number of molecules present in a given volume, at the same pressure and temperature, is approximately the same for all gases.

Since in Equation (39) there are twice as many molecules of hydrogen as of oxygen, the volume required to contain the hydrogen is twice as large as the volume containing the oxygen. The volume of the water vapor must also be twice as large as the volume of oxygen.

Thus, equations written in the molecular form show directly, by the coefficients, the volumes entering into the reaction.

151. Weight of Air Theoretically Required for Combustion per Pound of Coal. — The proportion of carbon, hydrogen, and sulphur present in the coal is obtained from the ultimate analysis, and the weight of air required to burn each is obtained as follows:

1. *Weight of air per pound of carbon for complete combustion:* The reaction equation is

1 atom carbon		1 molecule oxygen		1 molecule carbon dioxide	
C	+	O_2	=	CO_2	(40)
12	+	32	=	44	
1	+	$2\frac{2}{3}$	=	$3\frac{2}{3}$	

On a basis of weight, the above equation shows that 1 pound of carbon combines with $2\frac{2}{3}$ pounds of oxygen to form $3\frac{2}{3}$ pounds of carbon dioxide, and on a basis of volume, that carbon burned with one volume of oxygen produces one volume of carbon dioxide. Therefore, the volume of gas, before and after the reaction, is the same, provided the volume is measured at the same pressure and temperature in each case.

The equation also shows that 1 pound of carbon requires $2\frac{2}{3}$ pounds of oxygen for complete combustion; and therefore, since air is 23 per cent oxygen by weight, the weight of air required per pound of carbon burned equals $2.667 \div 0.23 = 11.57$ pounds.

The heat liberated is 14,600 B.t.u. per pound of carbon burned to carbon dioxide.

1*a*. *Weight of air per pound of carbon for incomplete combustion:* The oxygen being insufficient, there is an excess of carbon, and the reaction that occurs is

2 atoms carbon		1 molecule oxygen		2 molecules carbon monoxide	
$2C$	+	O_2	=	$2\,CO$	(41)
24	+	32	=	56	
1	+	$1\frac{1}{3}$	=	$2\frac{1}{3}$	

Equation (41) shows that $1\frac{1}{3}$ pounds of oxygen are required to burn 1 pound of carbon to carbon monoxide; therefore, the weight of air required per pound of carbon $= 1\frac{1}{3} \div 0.23 = 5.76$ pounds.

The heat liberated is 4380 B.t.u. per pound of carbon burned to carbon monoxide; consequently, the loss resulting from incomplete combustion $= 14{,}600 - 4380 = 10{,}220$ B.t.u. per pound of carbon burned to carbon monoxide.

2. *Weight of air per pound of hydrogen*: Article 150 shows that 1 pound of hydrogen unites with 8 pounds of oxygen to form water, and therefore requires $8 \div 0.23 = 34.8$ pounds of air per pound of hydrogen.

The heat liberated per pound of hydrogen, when burned to water, equals 62,000 B.t.u.

3. *Weight of air per pound of sulphur:* The combustion of sulphur occurs according to the following equation

1 *atom sulphur*		1 *molecule oxygen*		1 *molecule sulphur dioxide*	
S	+	O_2	=	SO_2 . . .	(42)
32	+	32	=	64	
1	+	1	=	2	

This equation shows that the weight of air required, per pound of sulphur burned to sulphur dioxide $= 1 \div 0.23 = 4.35$ pounds.

The heat evolved is 4000 B.t.u. per pound of sulphur burned to sulphur dioxide.

The sum of items 1, 2, and 3 gives the following equation for the weight of air per pound of coal:

$$\textit{Weight of air per pound of coal, lb.} = 11.6\,C + 34.8\left(H - \frac{O}{8}\right) + 4.35\,S \quad (43)$$

in which C, H, O, and S are the weights of carbon, hydrogen, oxygen, and sulphur per pound of coal, as shown by the ultimate analysis. As noted elsewhere, the "available" hydrogen, $H - \frac{O}{8}$, is less than the amount of hydrogen as shown by the analysis, on account of the appropriation by the oxygen of one-eighth its weight of hydrogen.

If combustion is incomplete, the air required for the proportion of the carbon burned to carbon monoxide must be multiplied by the weight of air required for incomplete combustion of carbon, as given by Equation (41), and the result added to Equation (43).

Example 19. — The ultimate analysis of the coal "as fired," from the test given in Example 20 and expressed in per cent, is: carbon, 83.68; hydrogen, 4.70; oxygen, 4.25; nitrogen, 1.61; sulphur, 0.73. Find the theoretical weight of air per pound of coal "as fired."

Solution. — Substituting the values given above in Equation (43).

$$\text{Weight of air per pound of coal} = 11.6\,C + 34.8\left(H - \frac{O}{8}\right) + 4.35\,S$$

$$= 11.6 \times .8368 + 34.8\left(.047 - \frac{.0425}{8}\right) + 4.35 \times .0073$$

$$= 11.19 \text{ lb.}$$

The weights and volumes of air and oxygen required to burn the common combustibles, together with the weights and volumes of the resulting products, are tabulated in Table 17

TABLE 17. — WEIGHT AND VOLUME OF OXYGEN AND AIR REQUIRED FOR COMBUSTION.

At 32° and 29.92 in. mercury

Combustible or Oxidizable Substance Column 1	Chemical Symbol	Reaction Equation	Product of Combustion	Weight required per Lb. of Column 1		Weight of resultants per Lb. of Column 1		Heat Value per Lb. of Column 1 B.t.u.
				Oxygen Lb.	Air = 4.32 × O Lb.	Nitrogen = 332 × O Lb.	Gaseous Product Lb.	
Carbon	C	$C + 2\,O = CO_2$	Carbon dioxide	2 667	11.52	8.85	12.52	14,600
Carbon	C	$C + O = CO$	Carbon monoxide	1.333	5.76	4.43	6.76	4,380
Carbon-monoxide	CO	$CO + O = CO_2$	Carbon dioxide	0.571	2.47	1.90	3.47	10,150
Hydrogen	H	$2\,H + O = H_2O$	Water	8	34.56	26.56	35.56	62,000
Methane	CH_4	$CH_4 + 4\,O = CO_2 + 2\,H_2O$	Carbon dioxide and water	4	17.28	13.28	18.28	23,550
Ethylene	C_2H_4	$C_2H_4 + 3\,O_2 = 2\,CO_2 + 2\,H_2O$	Carbon dioxide and water	3.43	14 81	11.38	15.81	1,591
Sulphur	S	$S + 2\,O = SO_2$	Sulphur dioxide	1	3.32	3.32	5.32	4,050

TABLE 17. (*Continued*)

Combustible or Oxidizable Substance Column 1	Volumes of Column 1 entering combination	Volumes of Oxygen Combining with previous column	Volumes of Product Formed	Volume required per Lb. of Column 1		Volume of Resultants per Lb. of Column 1		
				Oxygen Cu. Ft.	Air Cu. Ft.	Products of Combustion Cu. Ft.	Nitrogen = 3.782 × O Cu. Ft.	Gas = sum of two previous Columns Cu. Ft.
Carbon	1 C	2	$2\,CO_2$	29.89	143.10	29.89	112.98	142.87
Carbon	1 C	1	2 CO	14.95	71.55	29.89	56.49	86.38
Carbon-monoxide	2 CO	1	$2\,CO_2$	6.40	30.62	12.80	24.20	37.00
Hydrogen	2 H	1	$2\,H_2O$	89.66	429.07	179.32	339.09	518.41
Methane	$1\,CH_4$	4	$1\,CO_2$, $2\,H_2O$	44 83	204 97	67 34	169.55	236.89
Ethylene	$1\,C_2H_4$	3	$2\,CO_2$, $2\,H_2O$	38 80	181 31	42 84	145.81	187.65
Sulphur	1	2	$2\,SO_2$	11.21	53.71	11.21	42 39	53.60

152. Actual Weight of Air Required per Pound of Coal. — The weight of air which should be actually supplied to the fuel depends upon the method of supplying the air, the method of firing the fuel, and the furnace conditions. If the fuel is fired continuously, the air introduced at suitable places, and the opportunity afforded for mixing with the gases at a sufficiently high temperature before cooling against boiler surfaces occurs, 30 to 40 per cent of air in excess of that theoretically required is sufficient. *In general, 50 to 100 per cent excess is required, with the figures running to 200 per cent excess under poor furnace conditions.* The volume of air corresponding to its weight is dependent upon its temperature and pressure, as shown by Table 3, page 80, which covers the range of temperatures at normal atmospheric pressure. For other temperatures the volumes may be calculated by the method explained under Art. 71, page 82.

153. Flue Gases. — The products of combustion that pass away, through the flue or uptake connections, to the stack, are called flue gases. The composition of the flue gas depends on the material burned and the completeness of the combustion. *Flue gas from burning coal ordinarily contains carbon dioxide, carbon monoxide, nitrogen, oxygen, unburned hydrocarbons, carbon in suspension as smoke and ash dust, a small percentage of superheated steam, and some sulphur dioxide.* When combustion takes place under the most favorable conditions, there should be no carbon monoxide, unburned carbon, or hydrocarbons in the flue gas. Sulphur dioxide has a corrosive effect on the steel walls of the smoke flues in the presence of moisture resulting from condensation of steam.

154. Flue-gas Analysis. — *This analysis generally determines the proportion by volume of the carbon dioxide, carbon monoxide, and oxygen present in the products of combustion.* The method of making the analysis is to pass a small sample volume of the gases, usually 100 cubic centimeters (c.c.), through a series of receptacles in each of which a chemical reagent absorbs one of the principal constituents of the chimney gas. A typical analysis, taken from the gases passing the smoke box in a return-tubular boiler, gave 12.6 per cent CO_2, 5.7 per cent O_2, 0.0 per cent CO and, by difference, 81.7 per cent N_2. An additional test is sometimes made for unburned hydrocarbons, by passing that part of the sample remaining after removal of the CO_2, O_2, and CO into a tube into which oxygen is introduced. The reduction in volume resulting from ignition of the mixture shows the proportion of hydrogen or hydrocarbons present.

The analysis indicates the conditions under which combustion is taking place, and makes it possible to calculate the amount and distribution of the losses. Analyses are made for the following purposes:

1. To determine the best method of firing and handling the fire for the coal in use, with regard to coking, spreading, and leveling to prevent holes in the fuel bed.
2. To establish the best draft, considering thickness of fire and rate of combustion.
3. To discover air leaks in the setting.
4. To find amount of additional air required for complete combustion.
5. To obtain information bearing upon the proper design and construction of the furnace.

155. Sampling of Flue Gas. — Since the sample of gas drawn out for analysis should represent all the gas passing the point at which the sampling device is located, it is difficult to obtain a proper sample. A sampling tube, consisting of a pipe having a closed end and a series of perforations arranged in a lengthwise row, is sometimes used, in the passes of boilers and at various points along the gas stream, to collect the sample. The objections to this form of tube are that the holes may be filled by soot, and that those nearest

the suction end will furnish most gas. Besides, the velocity of the gas stream varies across its section, making it more difficult to obtain a true sample. These objections may be overcome to some extent by graduating the size of the holes.

The Power Test Committee of the A. S. M. E. recommends that: "the sample should be drawn from the region near the center of the main body of the escaping gases, using a sampling pipe not larger than ¼-inch gas pipe. The pipe should contain perforations extending the whole length of the part immersed and pointing toward the current of gas; the collective area of the perforations being less than the area of the pipe." The **Bureau of Mines** recommends a water-cooled tube or quartz tube, as preferable to a metal tube, to overcome any effect of the hot tube on the gas. An open-end tube located in the center of the gas stream gives results accurate within 0.5 per cent of the average composition, if the setting is fairly tight.

156. Location of Sampling Tube. — The location of the tube depends upon the information sought. If it is desired to find the conditions in the combustion space above the fuel bed, as for example in a B. and W. type boiler, the sampling pipe may be introduced from the side clean-out doors, between the tubes of the first pass. Care must be taken to prevent an excess of air entering around the sampling tube. *The usual location of the tube is near the boiler damper on the furnace side, where the effects of leakage of air into the setting will be shown.*

157. Collection of the Sample. — The flue gas is drawn and collected over water, by means of a water aspirator or some form of displacement apparatus. The water should be saturated with gas, to prevent absorption of CO_2 during the analysis. A brine solution absorbs CO_2 less readily than water, and may be used when it is necessary for the sample to stand for several hours before analysis. The gas should be analyzed immediately after being drawn, for best results. Conditions may change rapidly during combustion, and for this reason snap samples should be taken at short intervals. A continuous sample is necessary, in order to obtain an average that is of value.

158. Gas Analysis Apparatus. — There are two principal types of apparatus used in power plants for analyzing gases, the hand, or portable type, and the automatic-recording, stationary type. The **Orsat** is the most widely used hand apparatus. A description of this instrument, as given by the Bureau of Mines, follows:

"The essential parts of the apparatus, Fig. 86, are a 100-c.c. measuring burette graduated in 100 units with each unit sub-divided into fifths, four absorption pipettes, a leveling bottle, and a connection header; all of glass. The measuring burette is usually enclosed in a water jacket, which prevents sudden temperature changes while the analysis is being made. The pipettes are U-shaped glass vessels, and contain solutions for absorbing the

gas constituents. *The first contains caustic potash solution for absorbing the carbon dioxide. The second contains an alkaline solution of pyrogallic acid for absorbing oxygen, while the third and fourth contain an ammoniacal solution of cuprous chloride for absorbing the carbon monoxide.* The side of the pipette in which the absorption is to take place is filled with small glass tubes, which increases the surface of liquid in contact with the gas. The pyrogallic acid solution and the cuprous chloride solution absorb oxygen readily, and *air must be kept from coming in contact with them.* A rubber-bag is placed on the rear side of both these pipettes or a water-seal, consisting of a fourth pipette, is used. The gas is drawn into the burette and forced out of it by means of a **leveling bottle** attached to the lower end of the burette by a 3-foot rubber tube. The leveling bottle also permits regulation of the pressure on the gas. The header connecting the burette and the pipette is made of $\frac{1}{4}$-inch glass tubing. Each outlet is sometimes fitted with a ground-glass cock, but such cocks require constant attention to prevent them from sticking and leaking. **Pinch-cocks** on the rubber tubing are equally satisfactory."

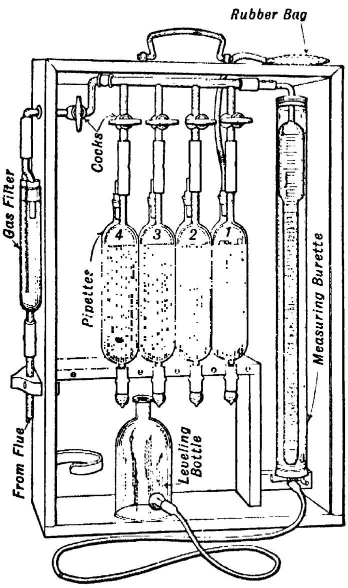

FIG. 86. — Orsat Apparatus.

The preparation of the solutions should be undertaken only by one who has the required knowledge. The solution for determination of CO_2 can be used for about 150 determinations.

Manipulation. — The manipulation of the apparatus may be briefly described as follows:

1. The apparatus is attached to the bottle containing the sample, and 102 c.c. of sample gas is drawn into the measuring burette by lowering the leveling bottle. The gas should be drawn through a gas filter placed in the tube furnishing the gas sample, in order to free the gas of mechanical impurities, such as carbon dust. The filter may consist of a glass tube, larger at the center than at the ends and loosely filled with mineral wool or, in some cases, with clean cotton waste.

2. The level of the water in the bottle is raised to a position opposite the 100 c.c. mark on the burette, and the three-way cock opened to the air to

relieve pressure on the gas. *All readings of the burette are taken with the levels of water in bottle and burette at same height.*

3. The cock to the first pipette where CO_2 is absorbed is opened and the water bottle raised, transferring the gas to the pipette. The gas is allowed to remain in the pipette several minutes and the solution is worked up and down, using the leveling bottle to assist the action. The gas is then transferred, by lowering the water bottle back to the burette for measurement, after which the gas is again transferred to the pipette and the process repeated until a constant reading is obtained for the CO_2. The same method is then used with the second and third pipettes. The difference in the readings of the burette at the start and finish, for each gas, gives the volume of gas absorbed. The process must be carried out in the order given; otherwise, the results will be incorrect.

The water and the solutions should not mix; to prevent this the movement of the leveling bottle should be stopped when either the water or the solution reaches the mark on the tube attached to the pipette.

Care must be taken, when transferring the sample to the apparatus, to fill the ends of the tubes connecting the Orsat and the sample-container with water, to prevent entrance of air.

159. Automatic Flue-gas Recorders. — These recorders determine only CO_2. In the intermittent recorder, which is essentially an automatic Orsat instrument, the operating power is obtained from water, piped to the device. The 100 c.c. sample is drawn from the pipe leading from the boiler, and transferred to a receiver containing the solution for absorbing CO_2. The gas sample passes through the solution to the space below a gas bell that is free to rise and fall. The volume of gas remaining after the absorption of the CO_2 determines the position of the bell, and a pen attached marks the position corresponding to the per cent of CO_2 on a clock-driven chart. The machine is arranged to analyze 6 or 7 samples per hour.

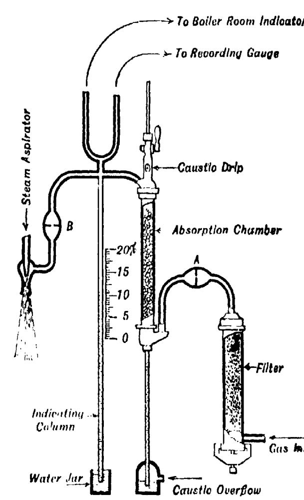

FIG. 87. — Principle of the Uehling Gas Composimeter.

The Uehling Composimeter is an example of the continuous type of meter. The principle of this meter may be explained by reference to Fig. 87. It consists of a filter, two orifices, *A* and *B*, an absorption chamber, and a steam aspirator. Gas is constantly drawn from the sample pipe

through the orifices, by the steam aspirator. The absorption of carbon dioxide in the absorption chamber reduces the volume of gas passing the second orifice, and increases the vacuum between the two orifices in proportion to the volume absorbed. A water column indicates the amount of vacuum and is generally graduated in per cent of CO_2. Suitable pipe connections are provided for recording instruments placed at a distance from the boiler room.

160. Deductions from the Gas Analysis. — For an intelligent judgment regarding the conditions of combustion, the readings of CO_2, O_2 and CO must be considered together.

Carbon dioxide, CO_2. — If the fuel were all carbon, perfect combustion would result in 20.9 per cent CO_2, since the volume of CO_2 formed is the same as the volume of oxygen used, as shown in Art. 151. This per cent is reduced, however, on account of the hydrogen in the fuel, because the nitrogen contained in the air used for burning the hydrogen remains in the flue gas and dilutes it. A further reduction occurs if hydrocarbon gas escapes without being burned, since it is accounted for as nitrogen by difference, if not separately determined. *The net result is that the sum of* CO_2 O, *and* CO *often equals* 18 *or* 19 *per cent.*

It is generally considered that 4 to 6 per cent CO_2 is very low, 10 to 12 per cent good, and above 12 per cent excellent. Low CO_2 is usually caused by a large excess of air, as shown by the curve, Fig. 88. It may be caused by an insufficient air supply, in which case the CO content is high, and the flue gas approaches a fuel gas in quality. Even with more than sufficient oxygen present, some CO may escape, if mixture with the gases does not occur at a suitable temperature for combustion before the gases are cooled by the heating surfaces. High CO_2, 12 to 15 per cent, is often accompanied by some loss due to the formation of CO. An analysis showing 0.1 to 0.2 per cent CO, with 12 to 13 per cent CO_2, indicates an excellent efficiency.

The maximum to which CO_2 can attain without formation of CO is dependent upon furnace conditions. Tests by the Bureau of Mines show the presence of CO with 8 to 10 per cent CO_2. "J. W. HAYS* does not consider that there is danger from CO until about 15 per cent CO_2 is reached." Tests have shown, however, 0.40 to 1.0 per cent CO for 13.8 to 15.6 per cent CO_2, under unusually good conditions. If the draft is proportional to the thickness of the fuel bed, high values of CO_2 can be produced, whether the rate of combustion, pounds per square foot of grate surface per hour, is high or low.

For perfect combustion, the reaction equation shows that the volume of carbon dioxide resulting from the combustion of carbon is the same as the volume of oxygen used. The CO_2 per pound of carbon burned should then

* Finding and Stopping Waste in Modern Boiler Rooms, H. S. Boiler Works.

be in the same proportion to the resulting flue gas as the oxygen is to the air supplied; that is, 20.89 per cent.

If 50 per cent excess air is supplied, then the 1.5 volumes of gas resulting contain 0.209 volumes CO_2, from which $\frac{0.209}{1.5} = 0.139$ or 13.9 per cent is CO_2. If 100 per cent excess air is supplied, the CO_2 in the flue gas is $\frac{0.209}{1.0 + 1.0} = 0.1045$ or 10.45 per cent.

The curve, Fig. 88, shows the values of CO_2 for corresponding values of excess air by volume, with carbon as fuel and perfect combustion.

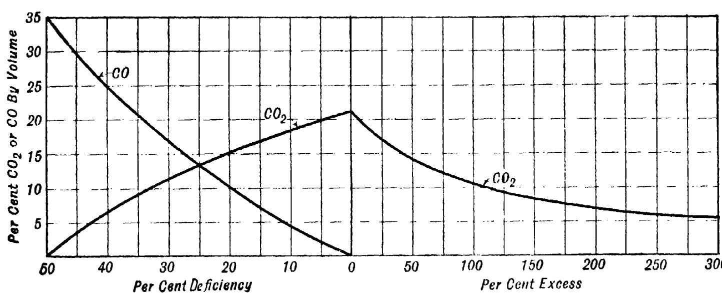

FIG. 88. — Effect of Air Supply on CO and CO_2 in the Flue Gas.

The excess air required to ensure combustion under good conditions is from 25 to 40 *per cent. Forty per cent excess air corresponds theoretically to a value of* 14.9 *per cent* CO_2.

Carbon monoxide, CO. — In general, the presence of carbon monoxide in flue gases indicates imperfect conditions for combustion. As A. BEMENT points out, this is a danger signal, not so much because of the loss it represents (never over 1 per cent of the heat value of the fuel) as because it is usually accompanied by the loss of unconsumed hydrogen and hydrocarbons.

The formation of carbon monoxide is due to:

1. Improper methods of firing.
2. Deficient air supply.
3. Improper mixing of the combustible gases with the air.
4. Low temperature of furnace, allowing gases to be cooled below ignition temperature.
5. Furnace not adapted to fuel to be burned, or responsible, on account of poor design, for (3) and (4).

Oxygen, O_2. — A high percentage of O_2 shows directly that a considerable gain in efficiency can be made by, (1) covering holes in the fire bed, (2)

stopping the entrance of air through the setting, or (3) adjusting the draft to the thickness of the fuel bed.

161. Combustion Indicators. — Combustion indicators, consisting of a special combination of draft gages, show, by variation from a reading that has been established for standard conditions, the necessity for attention to the fires. The Blonck boiler-efficiency meter, Fig. 89, is typical. It consists of *two draft gages, one connected between the ashpit and furnace, and the other between the furnace and the boiler side of the damper, in the breeching.* The lower gage contains *red oil* and shows the drop in pressure through the fuel bed; the upper gage contains *blue oil* and shows the drop in pressure between the furnace and damper. Sliding pointers mark the correct position of the fluid in the draft gages, and the movements of the fluid, to right or left from this position, give the firemen the information necessary for controlling the depth of fuel bed, air supply, and rate of combustion.

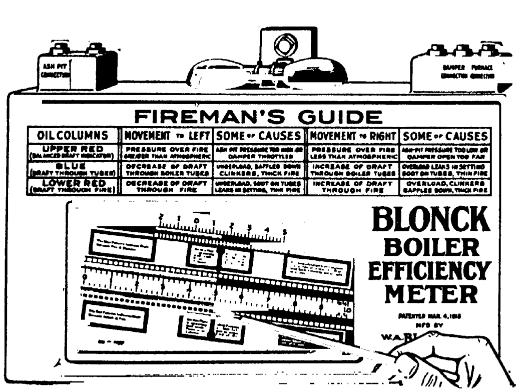

Fig. 89. — Blonck Combustion Indicator.

162. Weight of Air Supplied per Pound of Coal. — In most cases, the air supplied to burn the coal cannot be measured directly; but the amount of air can be calculated quite accurately, by means of an analysis of a representative sample of the flue gas, together with the proportion of carbon burned per pound of coal, as obtained by using the ultimate analyses of the coal and ash.

The calculation by this method is based upon the weight of nitrogen, as given by the flue-gas analysis. This nitrogen comes mostly from the air, while a small amount is supplied by the coal. *The nitrogen per pound of coal is determined from the flue-gas analysis, and the nitrogen from the coal is subtracted to give the nitrogen supplied by the air per pound of coal.* This result, divided by the proportion of nitrogen by weight in the air, will give the weight of air per pound of coal. The method of making the calculation follows:

The percentages by weight of carbon, hydrogen, oxygen and nitrogen, as shown by the ultimate analysis of the coal, are represented by their respective chemical symbols C, H, O and N.

The percentage by volume of the constituents of the flue gas, as shown by the flue-gas analysis, are represented by their respective chemical symbols CO_2, CO, O_2 and N_2.

1. *Weight of carbon burned*

$$\textit{per pound of coal fired, lb.} = \frac{\textit{Total carbon burned, lb.}}{\textit{Total coal fired, lb.}}$$

Using symbols this may be written

$$W_c = \frac{WC - W_aC_a}{W} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (44)$$

in which

W_c = weight of carbon burned per pound of coal fired, lb.
W = weight of coal fired, lb.
W_a = weight of ash, lb.
C = proportional weight of carbon per pound of coal by ultimate analysis.
C_a = proportional weight of carbon per pound of ash by ultimate analysis.

2. *Weight of nitrogen*

$$\textit{per pound of carbon, lb.} = \frac{28\ N_2}{12\ (CO_2 + CO)} \quad . \; . \; . \; . \quad (45)$$

in which 28 N_2 = relative weight of nitrogen in the flue gas.
12 $(CO_2 + CO)$ = relative weight of carbon in the flue gas.

3. *Total weight of nitrogen, W_n, per pound of coal fired is* found by combining Equation (44) and (45), thus giving

$$W_n = \frac{WC - W_aC_a}{W}\left[\frac{28\ N_2}{12\ (CO_2 + CO)}\right] \quad . \; . \; . \; . \quad (46)$$

The *net weight of nitrogen* supplied by the air per pound of coal burned equals W_n minus the weight of nitrogen in the coal, which is small and is usually neglected.

4. *Weight of dry air, A_f, supplied per pound of coal fired* is found by dividing Equation (46) by the proportional weight of nitrogen in the air, that is, $\frac{77}{100}$.

$$A_f = \frac{WC - W_aC_a}{0.77\ W}\left[\frac{28\ N_2}{12\ (CO_2 + CO)}\right] \quad . \; . \; . \; . \; . \quad (47)$$

If it is desired to find the weight of air supplied per pound of coal fired, uncorrected for loss through the grates, the factor W_aC_a drops from Equation (47), which can then be simplified to give the following equation:

$$A_f = 3.03\ C\left[\frac{N_2}{CO_2 + CO}\right] \quad . \; . \; . \; . \; . \; . \; . \; . \quad (48)$$

stopping the entrance of air through the setting, or (3) adjusting the draft to the thickness of the fuel bed.

161. Combustion Indicators. — Combustion indicators, consisting of a special combination of draft gages, show, by variation from a reading that has been established for standard conditions, the necessity for attention to the fires. The Blonck boiler-efficiency meter, Fig. 89, is typical. It consists of *two draft gages, one connected between the ashpit and furnace, and the other between the furnace and the boiler side of the damper, in the breeching.* The lower gage contains *red oil* and shows the drop in pressure through the fuel bed; the upper gage contains *blue oil* and shows the drop in pressure between the furnace and damper. Sliding pointers mark the correct position of the fluid in the draft gages, and the movements of the fluid, to right or left from this position, give the firemen the information necessary for controlling the depth of fuel bed, air supply, and rate of combustion.

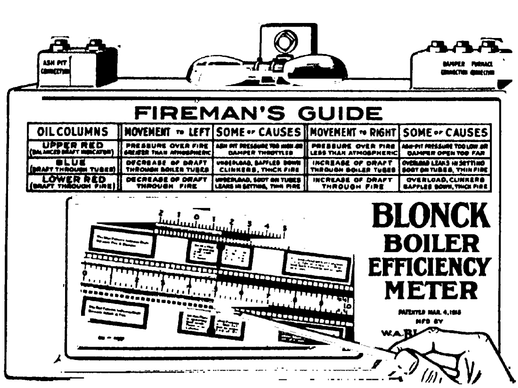

Fig. 89. — Blonck Combustion Indicator.

162. Weight of Air Supplied per Pound of Coal. — In most cases, the air supplied to burn the coal cannot be measured directly; but the amount of air can be calculated quite accurately, by means of an analysis of a representative sample of the flue gas, together with the proportion of carbon burned per pound of coal, as obtained by using the ultimate analyses of the coal and ash.

The calculation by this method is based upon the weight of nitrogen, as given by the flue-gas analysis. This nitrogen comes mostly from the air, while a small amount is supplied by the coal. *The nitrogen per pound of coal is determined from the flue-gas analysis, and the nitrogen from the coal is subtracted to give the nitrogen supplied by the air per pound of coal.* This result, divided by the proportion of nitrogen by weight in the air, will give the weight of air per pound of coal. The method of making the calculation follows:

The percentages by weight of carbon, hydrogen, oxygen and nitrogen, as shown by the ultimate analysis of the coal, are represented by their respective chemical symbols C, H, O and N.

Since the nitrogen in the flue-gas analysis is found by difference, $N_2 = 100 - CO_2 - CO - O_2$. Substituting this value of N_2 in Equation (49) and reducing, there results.

$$\textit{Weight of dry chimney gas per pound carbon, lb.} = \frac{4\,CO_2 + O_2 + 700}{3\,(CO_2 + CO)} \quad (50)$$

The weight of dry chimney gas per pound of coal fired is obtained by multiplying Equations (44) and (50) together, thus giving

$$W_d = \frac{WC - W_aC_a}{W}\left[\frac{4\,CO_2 + O_2 + 700}{3\,(CO_2 + CO)}\right] \quad . \quad . \quad . \quad . \quad . \quad (51)$$

in which W_d = the weight of dry chimney gas per pound of coal **fired.**

164. Total Weight of Gas per Pound of Coal. — This includes the weight of steam formed by combustion of hydrogen ($9\,H \div 100$), the weight of moisture shown directly by the analysis, and the weight of dry gas per pound of coal.

Example 21. — Using the data from Example 20, page 146, find: (*a*) The weight of dry chimney gases per pound of coal burned, (*b*) the total weight of gas per pound of coal.

Solution. — (*a*) The weight of dry chimney gases per pound of coal fired

$$= \frac{WC - W_aC_a}{W}\left[\frac{4\,CO_2 + O_2 + 700}{3\,(CO_2 + CO)}\right]$$

$$= \frac{10{,}970 \times 0.8368 - 670 \times 0.168}{10{,}970}\left[\frac{4 \times 14.3 + 3.0 + 700}{3(14.3 + 0.80)}\right]$$

$$= 13.86 \text{ lb.}$$

(*b*) The total weight of gas per pound of coal

$$= \text{Weight of steam} + \text{dry gas per pound of coal} + \text{moisture}$$
$$= 9 \times 0.047 + 13.86 + 0.022 = 14.3 \text{ lb.}$$

165. Excess Air. — The weight of excess air equals the difference between the weight of air shown by calculation from data of the ultimate analysis, Equation (43), and the weight found from the flue-gas analysis, Equation (48).

The excess air supplied is sometimes calculated by the use of the following formula:

$$\text{Per cent excess air} = \left[\frac{N_2}{N_2 - 3.782\,O_2} - 1\right] \times 100 \quad . \quad . \quad (52)$$

The values of N_2 and O_2 are taken from the gas analysis. This formula assumes that the oxygen present is in excess of that theoretically required, a condition that may not exist in reality.

Example 22. — Using the results of Examples 19 and 20 find the amount of excess air.

Solution. — From Example 19, the theoretical weight of air is 11.19 lb.
Actual weight of air, as calculated in Example 20, is 13.75 lb.
The weight of excess air = (13.75 − 11.19) = 2.56 lb.

Per cent excess air $\frac{2.56}{11.19} \times 100 = 22.9.$

166. Air Infiltration. — Special attention should be directed to the losses incurred by infiltration of air, in excess of that required for combustion. This air, passing through holes in the fuel bed and through holes or cracks in the furnace or boiler setting, carries away with it a large amount of heat, and reduces the overall efficiency of the boiler plant. This condition is revealed by the gas analysis, by the increase in the volume of oxygen, and the decrease in the volume of CO_2. The BUREAU OF MINES gives the following example:

		CO_2	O_2	CO
Rear of combustion chamber	Av.	14.5	3.3	0
Base of stack	Av.	10.7	8.1	0

167. Boiler Losses. — The heat losses which occur in the operation of boiler plants are included under the following headings:

1. Heat carried away in the dry chimney gases.
2. Evaporation of moisture formed by burning the hydrogen.
3. Evaporation of moisture in the fuel and air used.
4. Incomplete combustion of the fuel.
5. Unconsumed carbon in the ash.
6. Heating of moisture in the air.
7. Radiation and all other losses.

1. *Heat carried away in the dry chimney gases.* — The loss of heat in the chimney gases is usually the largest loss, amounting to 10 or 12 per cent of the heat generated by the boiler under the most favorable conditions of standard practice, and to 20 or 40 per cent where the supply of air is greatly in excess and there is leakage of air through the boiler setting. The greatest gain in efficiency can usually be made by reducing the air supply to the proper amount for the fuel used, by controlling the draft, by preventing holes in the fuel bed, and by preventing the entrance of air through breaks in the setting.

The heat carried away by the dry chimney gases is calculated by means of the following equation:

$$h_1 = W_g C_p (T - t) \qquad (53)$$

in which h_1 = B.t.u. lost per pound of fuel.
W_g = weight of dry chimney gas per pound of fuel, lb.
T = temperature of escaping gases, deg. fahr.
t = temperature of air entering furnace, deg. fahr.
C_p = mean specific heat of the dry gases = 0.24.

2. *Evaporation of moisture formed by burning hydrogen.* — This loss is unpreventable if there is hydrogen in the fuel. For anthracite coal the loss per pound of combustible is about 2.5 per cent of its heat value, and for bituminous coal it varies from 3 to 4.6 per cent. The water that is formed

by combustion passes away as steam superheated to the temperature of the flue gases. The heat loss is

$$h_2 = 9H\,[212 - t + 971.7 + C_p\,(T - 212)] \quad . \quad . \quad . \quad . \quad (54)$$

in which h_2 = B.t.u. loss per pound of fuel.

H = weight of hydrogen per pound of fuel, lb.

212 = temperature of evaporation, deg. fahr.

t = temperature of coal as fired, deg. fahr.

971.7 = latent heat at atmospheric pressure.

C_p = mean specific heat of superheated steam = 0.47.

Other symbols as in Equation (53).

3. *Evaporation of the moisture in the fuel.* — Moisture in the fuel reduces boiler efficiency, because of the loss of heat required to evaporate it. In hand-firing some fuels, less coal appears to be lost through the grates if the fuel is wet before firing. *Tests have shown that there is generally no gain in efficiency as a result of wetting the coal.* The possibility of gain in a particular case where considerable hydrocarbon escapes unburned, lies in the avidity with which the nascent oxygen dissociated from the steam unites with the escaping hydrocarbons. The hydrogen from the steam is burned and liberates the same amount of heat as that required to dissociate it.

The loss in heat is

$$h_3 = W\,[212 - t + 971.7 + C_p\,(T - 212)] \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

in which h_3 = B.t.u. per pound of fuel.

W = weight of moisture per pound of fuel fired, lb.

Other symbols as in Equation (54).

4. *Incomplete combustion.* — This term usually refers to the loss by incomplete combustion of carbon. The loss due to the escape of unburned hydrocarbons is included in the losses which are unaccounted for. The loss resulting from the formation of carbon-monoxide gas is generally small, and usually does not exceed 2 per cent of the heat value per pound of coal. The heat loss is

$$h_4 = \left[\frac{CO}{CO_2 - CO}\right](14{,}600 - 4{,}380)\,C \quad . \quad . \quad . \quad . \quad . \quad (56)$$

in which C = proportion of carbon in the fuel, and CO and CO_2 are percentages by volume, as from the gas analysis.

5. *Unconsumed carbon in the ash.* — This is due to the method of working the fire or to a grate that is not suited to the fuel. Under good conditions it should not amount to more than 4 per cent.

The loss may be computed thus:

$$h_5 = h_a \frac{W_a}{W} C_a \quad . \quad (57)$$

in which h_5 = loss in B.t.u. per pound of fuel.

h_a = heat value per pound of combustible in the dry refuse, B.t.u.

C_a = per cent of combustible in the dry refuse.

W_a = weight of ash and refuse, lb.

W = weight of fuel fired, lb.

h_a may be considered to be the same as for carbon, with sufficient accuracy for most purposes.

6. *Heating moisture in the air.* — This loss is small and is calculated thus:

$$h_6 = \text{B.t.u. loss per pound of fuel} = MC_p\,(T - t) \quad . \quad . \quad . \quad (58)$$

in which M = weight of moisture in air used per pound of fuel, lb.

= weight of moisture per pound of air × weight of air per pound fuel.

= [(relative humidity × weight in pounds per cubic foot of water vapor at temperature, t, of air entering the furnace × volume of one pound of dry air at t deg. fahr.) × (weight of dry air per pound of fuel)]

C_p = mean specific heat of superheated steam = 0.47.

T = temperature of escaping gases deg. fahr.

t = temperature of air entering the furnace deg. fahr.

7. *Radiation and all other losses.* — These losses are taken as the difference between the heat value per pound of fuel and the sum of the losses 1 to 6 inclusive. For well-designed boiler furnaces, these losses vary from 2 to 6 per cent.

$$h_7 = \text{B.t.u. per pound of fuel} - (h_1 + h_2 + h_3 + h_4 + h_5 + h_6). \quad . \quad (59)$$

Example 23. — Using the data, Example 20, page 146, compute the boiler losses and tabulate in form of a heat balance.

Solution. — Distribution of heat per pound of coal "as fired."

Equation	Name and Calculation of Loss	B.t.u.	Per Cent
	Calorific value of coal "as fired" (by calorimeter)	14,546	
	1. Heat absorbed by the boiler $0.7552 \times 14{,}546$	10,985	75.52
53	2. Heat carried away in dry chimney gases $13.86 \times 0.24\,(434 - 72)$	1,204	8.28
54	3. Evaporation of moisture from burning hydrogen $9 \times 0.047\,[212 - 72 + 971.7 + 0.47\,(434 - 212)]$	514	3.54
55	4. Evaporation of moisture in the fuel fired $0.0221\,[212 - 72 + 971.7 + 0.47\,(434 - 212)]$	27	0.19
56	5. Incomplete combustion $0.83 \times (14{,}600 - 4380) \times \frac{0.8}{14.3 + 0.8}$	447	3.07
57	6. Unconsumed carbon in the ash $14{,}600 \times \frac{670 \times 0.20}{10.970}$	178	1.22
58	7. Superheating water vapor in the air $0.80 \times 0.0012 \times 13.14 \times 13.57 \times 0.47\,(434 - 72)$	29	0.20
59	8. Radiation and all other losses (by difference)	1,172	7.98
	Total losses	14,546	100.00

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Fundamentals of Combustion and Air Supply, AMERICAN ARCH Co.
Coal and its Combustion, COSGROVE.

REVIEW QUESTIONS AND PROBLEMS

1. Define: (*a*) combustion, (*b*) combustible.

2. Describe the changes which take place during combustion of a solid fuel.

3. Why is it more convenient to use the molecular form of chemical equation for combustion than the atomic form?

4. A Montana coal "as fired" has the following ultimate analysis, expressed in per cent: sulphur, 7.69; hydrogen, 3.16; carbon, 44.16; nitrogen, 0.49; oxygen, 6.73; ash, 37.77. Find the theoretical amount of air required per pound of coal.

5. In Problem 4, what would be the weight of air required, provided 50 per cent excess air were used? What effect would this excess air have on the percentage of carbon dioxide in the flue gas?

6. The ultimate analysis of a Kentucky coal "as fired," expressed in per cent, is as follows: sulphur, 1.22; hydrogen, 5.43; carbon, 77.37; nitrogen, 1.83; oxygen, 9.76; ash, 4.39. Find the weight of air theoretically required to burn a pound of the coal. Would this be sufficient under actual conditions?

7. What is the function of the air supplied for combustion?

8. Explain the meaning of all factors and symbols used in Equation (43).

9. What is a flue-gas analysis? How is it made, and what does it show regarding combustion?

10. Explain the principle of operation of a continuous type of carbon-dioxide meter.

11. During the test of a 508 horsepower B. and W. boiler, the following data were taken: weight of coal fired per hour, 21,954 lb.; weight of ashes per hour, 4087 lb.; carbon in the ash, 22.84 per cent.

Ultimate analysis is of coal "as fired," expressed in per cent:

Carbon	60.31	Nitrogen	1.13
Hydrogen	4.06	Sulphur	5.23
Oxygen	6.71	Ash	22.56

Flue-gas analysis, expressed in per cent:

Carbon dioxide	10.69	Oxygen	8.23
Carbon monoxide	0.23	Nitrogen (by difference)	80.85

Find the weight of dry chimney gases per pound of coal fired. What must be added to the weight of the dry gases to obtain the total weight of chimney gases?

12. What is the effect upon the flue-gas analysis of air leaking into a boiler setting? How can this leakage be determined by means of flue-gas analyses?

13. What is shown by a boiler heat balance? Of what assistance is such a heat balance in correcting improper furnace conditions?

CHAPTER VIII

SMOKE PREVENTION, FURNACES AND STOKERS

168. Foreword. — Within the past few years, great advances have been made in the methods used to prevent smoke in power plants burning coal. These improvements have been brought about by campaigns against smoke-producing plants, and by more stringent enforcement of city ordinances against smoke.

The better grades of fuel, which contain small amounts of volatile matter, may easily be burned without smoke; but with the poorer grades of fuel, high in volatile matter, the prevention of smoke is more difficult, and careful attention must be given to furnace conditions.

The actual loss of heat from smoke alone does not exceed $1\frac{3}{4}$ per cent of the heat value of the fuel. The presence of dense smoke, however, indicates poor furnace conditions, which may cause losses amounting to 20 per cent. Such losses materially reduce the efficiency and capacity of a boiler plant. On the other hand, a smokeless stack does not necessarily mean high efficiency in the boiler room; it may indicate too large an excess of air and the accompanying losses. A small amount of smoke may accompany the best of furnace conditions.

169. Smoke and Its Cause. — Smoke consists of water vapor and gaseous products of combustion, colored with fine particles of carbon or soot, and unburned vapors of the tarry constituents of fuel.

Smoke is produced if the air supply is insufficient; if the air and gases are not intimately mixed; if the temperature is not high enough to ignite the carbon; or if there is not sufficient time to burn the gases completely, before they have been chilled below the temperature of ignition by contact with the cooler surfaces of the boiler.

To prevent smoke, the escape of the unburned smoke-making constituents must be prevented, and the conditions necessary for perfect combustion, Art. 145, must be maintained. The volume of the combustion chamber must be sufficient to permit the proper mixing of the air and gases before they are cooled below the ignition temperature, and to assure their burning before reaching the boiler heating surface.

Coal that has a high volatile-matter content is not necessarily a smoke producer, the amount of smoke depending upon the constituents forming the volatile matter. A high hydrocarbon content in the volatile matter generally indicates that the coal will produce a large amount of smoke.

A coal which has a large amount of tarry vapors will produce smoke, unless ample time is given in which to burn them.

A knowledge of firing methods is essential to a proper understanding of the methods employed to prevent smoke. The methods used to charge coal are (1) hand stoking, and (2) mechanical stoking.

170. Grates for Hand Stoking. — The purpose of the grate is to support the fuel bed, at the same time admitting air for combustion. It should be of such a form that it will be kept uniformly cool by the inflowing air, and should permit ashes to pass freely. The width of the air spaces should be as great as possible without permitting coal to pass through. This width depends upon the kind and size of coal used, and ranges from $\frac{3}{16}$ to $\frac{5}{16}$ of an inch. *The total area of the openings for air, when burning coal, varies from* 30 *to* 50 *per cent of the total grate area.*

The type of grate to be used with a hand-fired furnace is controlled by the type of boiler and kind of fuel used. With small vertical boilers, a **stationary circular grate,** Fig. 90, is used; it is made of a number of cast-

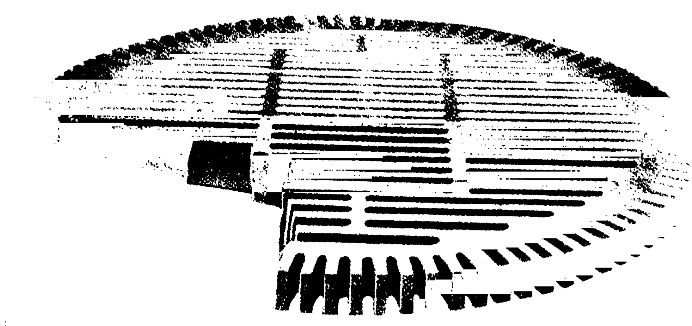
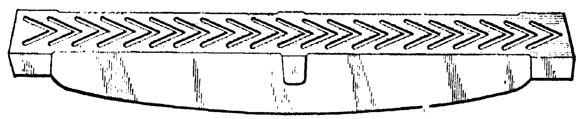


Fig. 90. — Circular Stationary Grate.

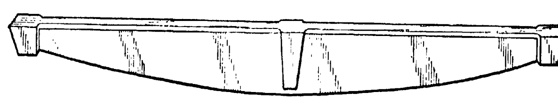
iron pieces fitted to form a circle, and is supported by a suitable ring, Fig. 28, page 35. With large vertical boilers, **rocking** and **shaking grates** of the circular form are used, while horizontal boilers often use **stationary grates** with the grate surface formed by a number of grate bars, placed side by side and held in place by their own weight. Lugs at each end of the grate bars rest upon **bearer bars** fastened to the walls of the setting. The **common grate bar,** Fig. 91, is made about 3 inches deep at its center, to give strength and thus prevent sagging under the weight of coal when hot. The width of each bar varies from $\frac{3}{4}$ inch at the top to $\frac{3}{8}$ inch at the bottom, to allow the ashes to drop through easily.

The tupper and herring-bone grate bar is made with side flanges, which stiffen the grate and reduce the liability of warping. Each grate bar is about 6 inches wide and has openings of the shape shown.

The length of each grate bar is about 3 feet. The total length of the grate depends upon the type of fuel and size of boiler. For easy firing, the length should not be over 6 feet. The rear end of the grate is ordinarily from 3 to 5 inches lower than the front end. A solid plate known as



Herringbone grate bar—6 in. wide



Common grate bar—1½ in. wide

FIG. 91. — Types of Stationary Grate Bars.

the **dead plate** is often used, with coking coal, to support the front end of the grate bars and to hold the green fuel until the volatile gases are distilled off.

171. Shaking Grates. — With stationary grates, the fire is not easily cleaned, combustion is hindered and the fire is sluggish, unless the air

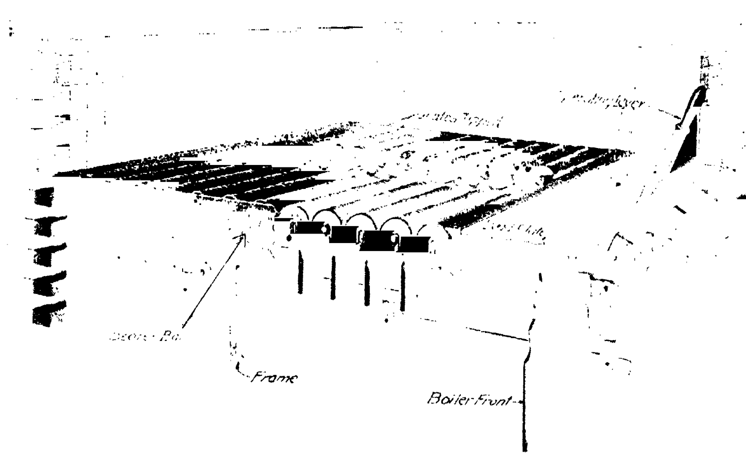


FIG. 92. — Hand-operated Shaking Grate.

spaces are kept free of clinker and ashes. Frequent cleaning is wasteful of coal, and requires the firing doors to be open. This admits a large

quantity of excess air and consequently lowers the temperature of the furnace. The shaking grate, Fig. 92, permits removing the ash and refuse without opening the firing doors and also reduces the amount of labor required. At each end of the grate bars, which run at right angles to the length of the furnace, are trunnions resting in slots in the side **bearer bars.** Ordinarily the grate surface is 2 or 3 bars wide and a varying number of bars in depth. The side bearer bars are supported by a frame, which rests upon the ashpit floor. The rear and front grate bars are stationary, and the lower ends of the movable grate bars are joined in groups. Each group is connected to a lever attached to the shaking handle. Movement of the handle tips the grate bars about the lugs resting in the bearer bars, and thus allows the loose ash to fall into the ashpit.

172. Tools Used to Handle Fires. — The tools required to handle the fire, in a furnace fired by hand, are the **hoe,** the **slice bar** and the **rake.** The hoe and the slice bar are used principally to clean the fire and break up the clinker; the rake is used to level the fuel bed. A **lazy bar** is often used to support the tools while working; it is laid from the hinge bracket to the latch lug, and the tools rest upon it. Light, yet strong, tools may be made by welding handles and fire ends to 1- or $1\frac{1}{4}$-inch pipe, making the total length about 8 feet.

173. Methods of Firing Coal by Hand. — Coal is charged into hand-fired furnaces by the **alternate, spreading, coking,** and **ribbon methods.**

When the *alternate method* is used, coal is fired on one side of the furnace while the other is burning brightly. The volatile gases that are distilled off are mixed with air admitted over the fire and are burned in the presence of an intense heat from the bright side of the furnace.

With the *spreading method*, coal is spread in a thin layer over the grate at each firing. It is customary to commence at the bridge wall and work toward the firing door. This requires the frequent firing of small quantities of coal.

With the *coking method*, coal is charged on the dead plate or at the front of the fire, in considerable depth, and allowed to coke, after which the coke is pushed toward the bridge wall and spread evenly over the grate, before more coal is fired. The hot gases distilled during the coking period must pass over the burning fuel in the rear, and are burned.

By the *ribbon method*, coal is fired in narrow alternate strips, which extend the full length of the grate.

Of these four methods, tests have shown that the ribbon method of firing, with the coal fired in small amounts, gives the highest efficiency and practically no smoke; that the spreading method gives the lowest efficiency and produces the most smoke; and that the coking and alternate methods give nearly the same result and are better than the spreading method.

Coal is fired intermittently into hand-fired furnaces, and careful firing is required to prevent smoke. The results obtained depend upon the human element and generally are not consistent. The fires are liable to be carried too thick, with the result that the air supply is insufficient. As a general rule, a 9-inch fire is sufficiently thick, while tests by the BUREAU OF MINES show that *thinner fires* than this, *with the coal fired often and in small amounts, give the best results.*

174. Smoke Prevention, Hand-fired Furnaces. — The methods of hand firing which produce the least smoke approach the method of firing used with mechanical stokers. A modification of the shaking grate, known as a **hand-operated stoker,** is sometimes used effectively; the grate surface is formed as in an ordinary shaking grate, but is given a greater slope from front to rear, with a dump plate at the bottom of the slope and a coking plate at the top. The principal methods of smoke prevention in hand-fired furnaces are those which depend upon furnace construction to ensure thorough mixing of air and gases. Special baffle walls of the inclined type, Fig. 93, and furnaces of the Dutch-oven, Chicago-setting and Downdraft types are examples of the methods used.

The **Dutch-oven furnace,** Fig. 102, page 165, consists of an extension of the common type of setting, bringing the furnace out in front of the boiler. The hot gases thus have a greater distance to travel before coming in contact with the cooler boiler and consequently have more time in which to burn. The **coking arch** above the grate absorbs heat from the burning coal, becomes white hot, radiates heat back upon the coal, and thus assists in securing better combustion. The chief defect of the Dutch oven is too rapid distillation of the volatile gases, without sufficient air supply to properly burn them.

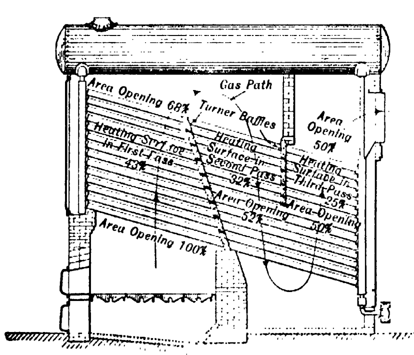


FIG. 93. — Inclined Baffle Walls.

The **Wing-wall Chicago setting** is used to burn Illinois bituminous coal. It differs from the ordinary boiler setting in that, directly back of the bridge wall, there is a wing-shaped firebrick wall, which changes the course of the furnace gases and produces better mixing of the air and gases. Like most hand-fired furnaces, it requires careful manipulation.

The **Downdraft furnace,** Fig. 94, has an upper and a lower grate. The upper grate is formed by a series of pipes through which water from

the boiler circulates, while the lower grate is formed by common grate-bars. Coal is thrown upon the upper grate, where it burns, and incandescent fuel, falling through to the lower grate, keeps a bright fire there.

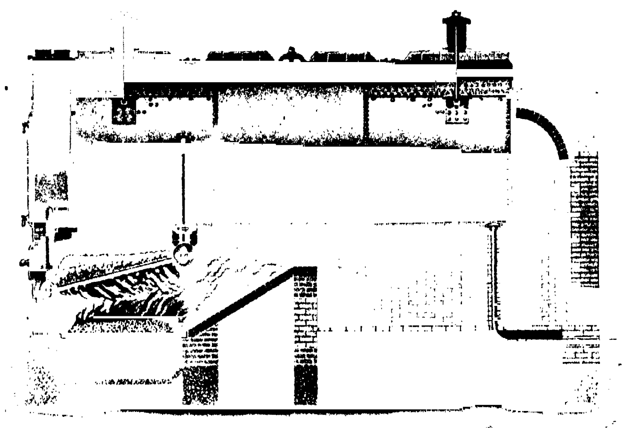


FIG. 94. — Vogt Downdraft Furnace.

Air admitted above the upper grate mixes with the distilled gases from the coking coal and passes down through the coal on the upper grate. These gases are mixed with additional air admitted through the lower grate doors, and are completely burned by the heat from the incandescent fuel on the lower grate. *This gives practically smokeless combustion.*

Steam jets are often used for inducing air and mixing it with the gases. The jets are placed above the grate and at the front end of the furnace, and are operated during the firing period and for a short time after firing.

A form of setting known as the **Kilgour setting** has been used successfully for prevention of smoke. It makes use of steam jets located on the bridge wall, and draws heated air through a passage in the bridge wall at the time of firing. The steam, air, and distilled gases are then directed through a short passage lined with firebrick, which becomes nearly white hot under ordinary conditions of operation.

175. Smoke Prevention by Mechanical Firing Methods. — The best method of preventing smoke in coal-fired furnaces is to use mechanical stokers. Their use presents the following **advantages**:

1. Reduces amount of labor required in the boiler room of large power plants. With stokers, a man can care for 5000 to 7000 boiler horsepower and, in extreme cases, as high as 14,000 boiler horsepower. With hand firing, a man is required for each 500 boiler horsepower.

2. Permits the burning of poorer grades of fuel.
3. Feeds the coal at a uniform rate and thereby maintains better furnace conditions.
4. Avoids excessive admission of air.
5. Saves labor of handling ashes and is self-cleaning.

The **disadvantages** are cost of operation and repairs, resulting from high furnace temperatures. As ordinarily operated, about 6 per cent of the steam generated is required to operate the stokers.

To obtain satisfactory results in the operation of stokers, intelligence and good judgment are required. Stokers are too often neglected, and therefore fail to give satisfaction.

In general, the simpler the stoker the better are the results obtained. Stokers are in common use in all power plants of 500 boiler horsepower or more, and in many plants of 200 boiler horsepower. Their use in small plants is increasing.

176. Classification of Stokers. — Stokers may be divided into three general classes:

1. Traveling, or chain grate.
2. Overfeed.
3. Underfeed.

The names of a few of the typical makes belonging to each class follow:

Chain-grate		Babcock and Wilcox Green Harrington Illinois Westinghouse Brady
Overfeed	Front-overfeed	Roney Wilkinson
	Side-overfeed	Murphy Detroit
Underfeed	Surface of fuel bed nearly flat	Jones American Type E
	Surface of fuel bed inclined at a considerable angle.	Riley Taylor Westinghouse

177. Traveling, or Chain-grate. — The grate surface of this type of stoker consists of an endless traveling chain made up of a large number of links, upon which the coal is fired at the front end and from which the ashes are discharged at the rear. The chain travels through the fur-

nace at a uniform rate and does not agitate the fuel bed. As usually set, the surface of the grate is horizontal; some makes, however, have a slight

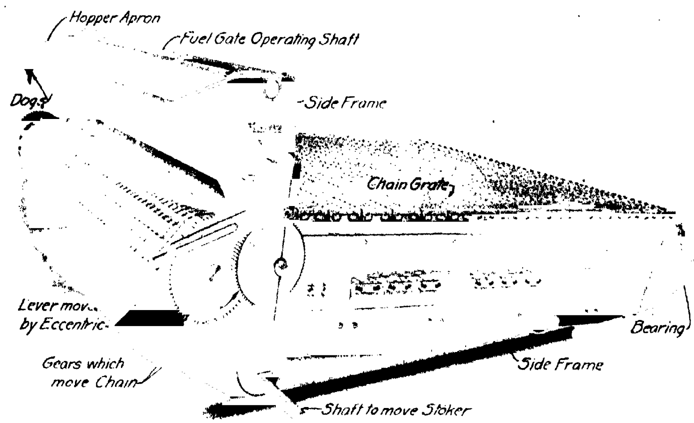


FIG. 95. — Green Type K Chain Grate Assembly.

incline toward the rear. The various makes differ mainly in points of design.

An assembly of the grate and frame is shown in Fig. 95, and the frame without the chain is illustrated in Fig. 96. The **frame** consists of cast-iron **side frames** joined by **steel cross girders** and shaped at the front

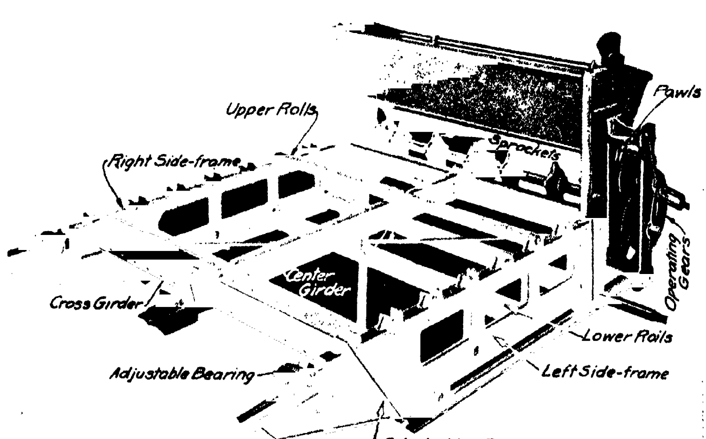


FIG. 96. — Assembly of Chain Grate Frame from Furnace End.

end to form the sides of the coal hopper. The chain rests on top and bottom **cross rolls,** which are supported by the side frames. The **front sprocket shaft** is supported in bearings fixed to the side frames, while the rear sprocket-shaft bearings are made adjustable, to permit changing the tension of the chain. Ordinarily the rear sprockets are replaced by drums.

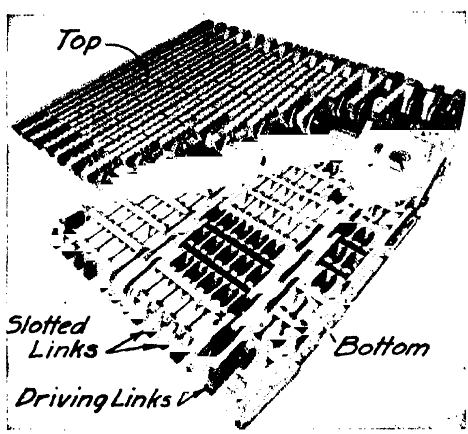


FIG. 97. — Green Chain Grate.

The **links** of the chain are held together by oval bars, with the **driving links** riding over the sprocket teeth centrally and carrying all tension. The remaining links are slotted for easy removal and do no driving. A top and bottom view of the chain is shown in Fig. 97. It is driven through a system of **spur gears,** which drive the front sprocket shaft from both sides. The spur gears are driven by a **pawl** which is attached to an arm connected to an eccentric rod, having an **eccentric** located on a shaft at the front of the boiler or below the boiler room floor. The eccentric shaft is generally driven by a belt from a small steam engine or electric motor. The position of the eccentric rod on the arm is adjustable, to permit changing the rate of feed; the longer the arm is made the slower will be the feed.

Ventilated cooling plates are sometimes placed in the side frame above the top of the grate, to prevent overheating of the side frames. **Adjustable flanges** on the **ledge plates,** which are built into the side walls of the setting, fit against the side of the upper chain, preventing air leakage between the chain and walls and presenting a durable surface to the moving fuel.

The **front apron** of the hopper is made of steel plate and can be dropped to a horizontal position by releasing the **holding dogs.** The back wall of the hopper is formed by a **fuel gate,** which is adjustable in height. This gate has a firebrick lining on the fire side, and renewable **iron shoes** on the hopper side. The depth of fuel on the grate is regulated by raising or lowering the gate by means of a worm and wheel. Leakage of air into the furnace through the rear is prevented by **side dampers** attached to the side walls. These dampers bear against projections on the side frame and make a tight joint at this point. A steel plate, Fig. 98, con-

necting the side frames, below the lower chain, prevents air leakage at the rear. The whole frame is mounted on wheels which rest on a track, thus permitting withdrawal from the furnace when making adjustments.

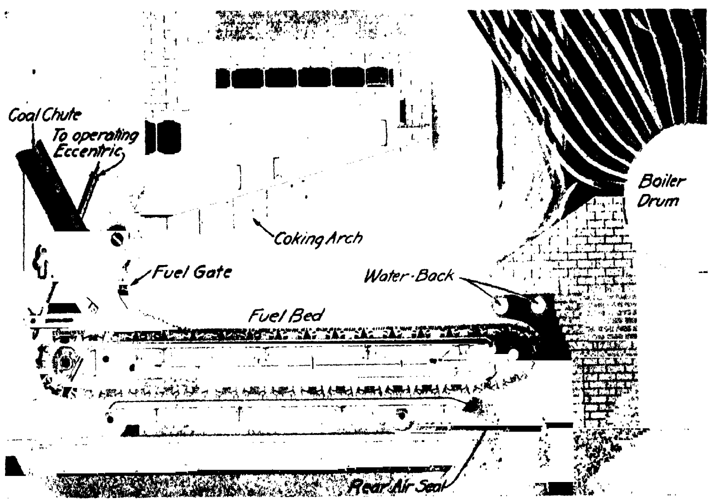


FIG. 98. — Green Chain Grate under Stirling Boiler.

Coal is fed by gravity from the hopper, Fig 98, to the moving chain grate. It passes under the fuel gate and is ignited, by the furnace fire, when under a **flat** or **sloping coking arch** which extends over the entire width of the furnace at the front end and *assists in the combustion of the fuel. The speed of the chain,* considering depth of fuel and draft, *is such that combustion will be complete by the time the chain passes the rear sprocket and the refuse falls to the ash pit below.*

The fuel bed at the rear end of the grate is compressed by coming into contact with an overhanging bridge wall. When this wall is built of fire-brick, incandescent ash adheres to it, the opening for ash discharge becomes closed, and the overhang burns off. To prevent this, a **water back,** Fig. 98, consisting of a pipe or pipes spanning the rear of the furnace, is used. Water passing to the boiler circulates through the pipes and keeps them cool. A cast-iron bridge wall, to which clinker does not adhere and which admits air over the fuel at the crest of the bridge wall, is sometimes used to replace the water back. It is claimed that the efficiency of the furnace is thus increased.

This type of stoker is one of the most common. It is adapted to free-burning coal that is high in volatile matter, but does not give as satisfactory operation with the high-carbon coking coals of the Appalachian

field. When intelligently operated, this stoker will produce little smoke and will handle small sizes of coal successfully. *The type and shape of the ignition arch are important.*

For best operation, the depth of fuel bed should not be over 6 or 8 inches. Coal should not be permitted to burn in the ashpit, and the grate bars should be cold when re-entering the furnace. A combustion rate of 24 pounds of coal per square foot of grate surface per hour may be used with satisfactory results, and higher rates for the peak load may be obtained with increased draft and depth of fuel.

178. Front-overfeed Stoker. — *This type of stoker has an inclined grate surface. Coal is fed on the grate at the top, and the ashes are discharged at the bottom.* Figure 99 shows a typical stoker of this type, as applied to a Vogt water-tube boiler.

The Westinghouse-Roney front-overfeed stoker, shown in Fig. 100, has a fuel hopper located at the front of the boiler and extending the

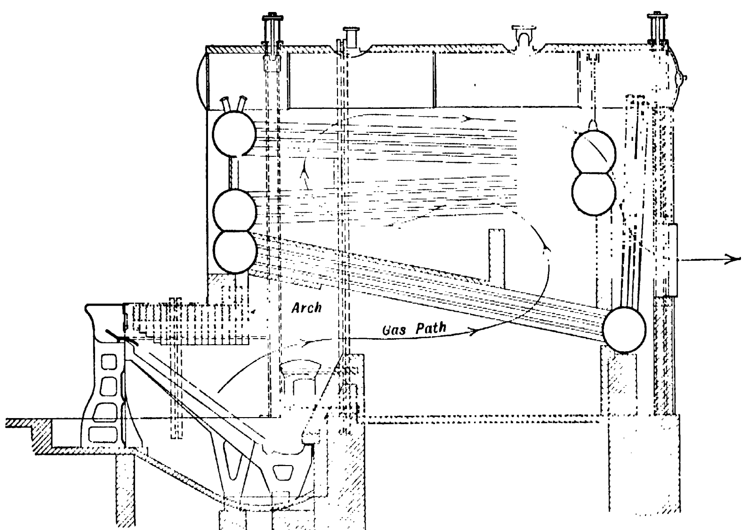


FIG. 99. — Westinghouse-Roney Stoker Applied to Vogt Water-tube Boiler.

width of the furnace. **Rocking grate bars,** Fig. 101, are inclined from front to rear and are supported at each end by bearer bars. The **main operating shaft** runs horizontally along the stoker front beneath the hopper. Keyed to this shaft is an eccentric which, by means of an arm cast integral with one-half of the eccentric strap, imparts an oscillatory motion to the agitator. This **agitator** swings freely on a hopper shaft, which has bearings at each end of the hopper. Connecting rods, by which the grate bars are

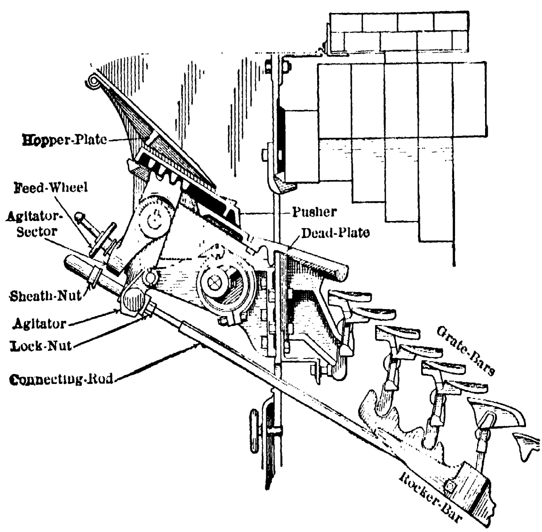


FIG. 100a. — Feed Mechanism, Roney Stoker.

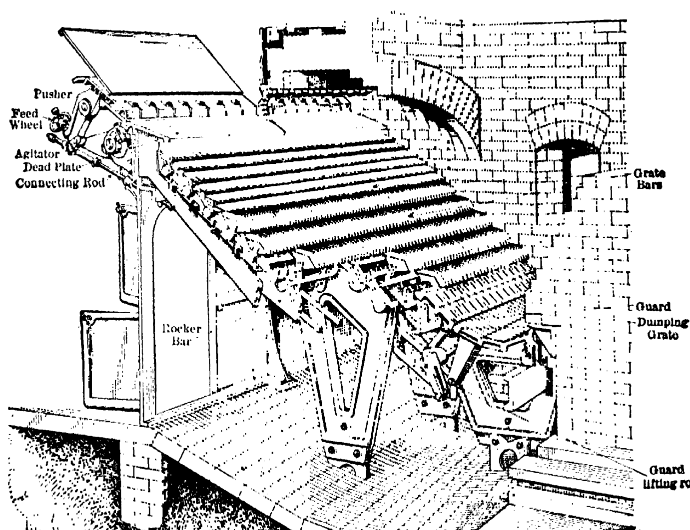


FIG. 100b. — Roney Stoker — Assembly.

rocked, are fastened to the agitator and the **rocker bar,** in which a projection on the grate bars rests. At each end of the hopper shaft, a **toothed sector** is keyed; it projects through a slot in the **pusher plate,** at the bottom of the hopper, and engages a series of **rack teeth** on the under side of the **pusher** which rests on the pusher plate and is operated by the toothed sector. To one of these sectors an offset arm is cast, the outer end overhanging the agitator. *A long bolt, attached by a swivel connection to the agitator, passes loosely through the offset arm of the sector, and, by means of a handwheel on the bolt, forms a means of adjusting the travel of the pusher, by allowing lost motion between the handwheel and the sector.* With the

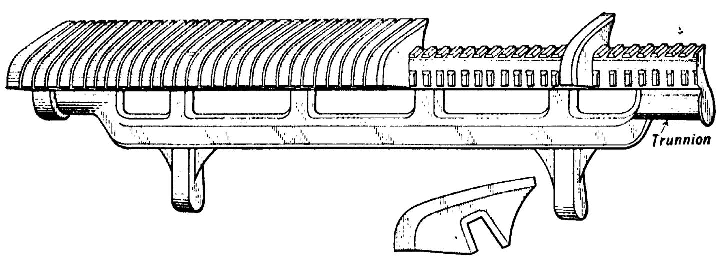


FIG. 101. — Section Grate Bar — Roney Stoker.

handwheel set up tightly, the agitator, through the hopper shaft and sectors, imparts maximum motion to the pusher. If the feed wheel is slacked off, so that there is lost motion equal to the arc described by the agitator, at the point where the bolt is attached, the pusher will not be moved.

A dump plate, made into several sections, for convenience in handling, is located at the bottom of the grate bars. At the top of the grate bars is a **dead plate** extending a short distance into the furnace, below the **combustion arch,** which extends over the width of the grate. A lug on each grate bar rests in a rocker bar, driven by the connecting rod attached to the agitator. *The rocker bar rocks the grate bars, on their trunnions in the side bearers, through an arc of* 30 *degrees, giving them first a horizontal and then an inclined position.* The amount of motion given the rocker bar is regulated by changing the position of the sheath and lock nuts, to increase or decrease the lost motion.

When the pusher is drawn back by the sector, coal from the hopper settles down in front of it and, as it advances, the coal is pushed on to the dead plate, where it remains a sufficient length of time for the volatile gases to be distilled from the coal by the radiant heat from the combustion arch. These gases are burned in passing to the rear over the burning fuel on the inclined grate. *The rocking motion of the grate bars, assisted by the force of gravity, causes the coal to move down the incline, and it is burned*

in traveling to the dump plate at the bottom, where the ashes collect and are dumped by hand as occasion demands. At the bottom of the incline a **guard plate** can be raised to hold the coal on the grate when lowering the dump plate.

Air for combustion is supplied by natural draft from below the grates. Steam jets at the front of the combustion arch are sometimes used to assist in mixing the air and gases.

This type of stoker can be readily forced beyond its rated capacity to higher rates of combustion, and if installed in a properly designed furnace

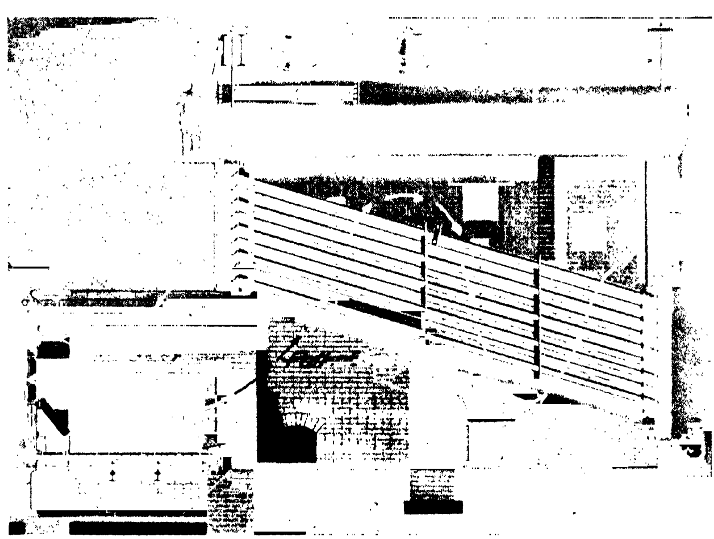


FIG. 102. — Murphy Stoker as applied to B. & W. Boiler.

will not produce much smoke. The rate of combustion can be as high as 40 pounds of coal per square foot of grate surface per hour, but for best operation it should be about 20 pounds per square foot per hour. The depth of the fuel should not be over 7 inches.

179. Side-overfeed Stokers. — This type of stoker is generally located in a Dutch-oven furnace, as shown in Fig. 102. *The grate bars are arranged to form a V-shaped inclined fuel bed which is agitated by the movable grate bars.*

The Murphy stoker, Fig. 103, which is representative of this type, has two fuel hoppers, one on each side of the furnace, extending from front to rear. The inner wall of the hopper is formed by an **arch plate** supported front and rear by an upright frame. Ribs, forming passages through which heated air from the air space above the arch is admitted for com-

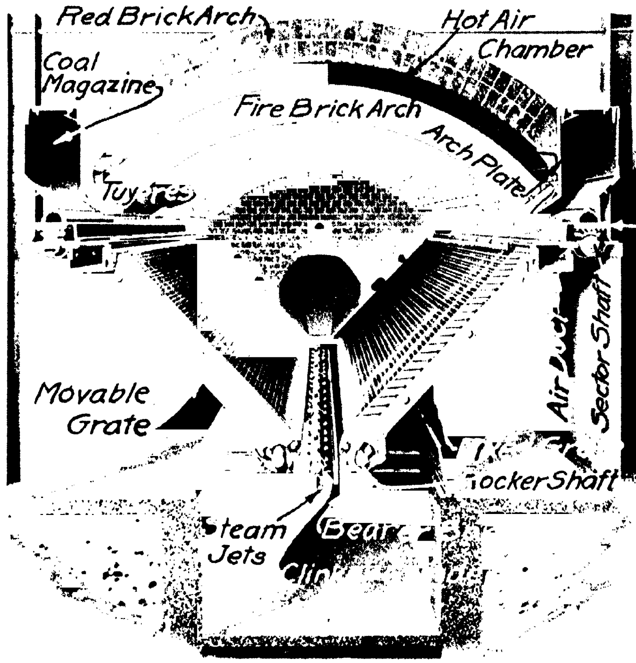


FIG. 103. — Murphy Stoker looking toward the Front End.

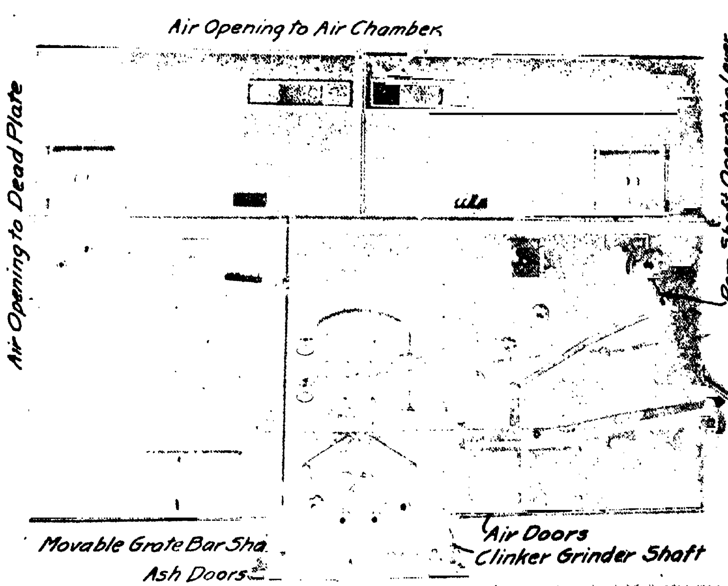


FIG. 104. — Front View of Murphy Stoker.

bustion, are cast on the lower end of the arch plate. At the bottom of the hopper is a built-up iron and steel plate, called the coking plate. Below this is an air duct through which air circulates and keeps the coking plate from becoming overheated. The pusher rests and slides upon the coking plate, as it is operated by a rack and sector. The sector is driven by a shaft, running below the hopper to the front of the furnace, Fig. 104, and is there connected to a driving arm.

The grate bars, Fig. 105, *are made in pairs, one fixed and one movable.* The fixed grate bars are ribbed to prevent the entrance of too much air from below the grate, and also to prevent excessive sifting of fine coal. Their upper ends rest against the coking plate and the lower ends are supported by the **grate bearer.** The movable grate bars are pivoted at their upper ends, and their lower ends are moved by the rocker shaft, first above and then below the stationary grate bars.

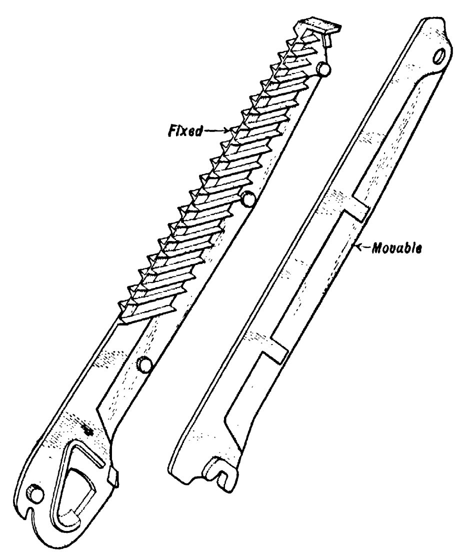


Fig. 105. — Grate Bars for Murphy Stoker.

The grate bearer is located at the bottom of the V and, besides supporting the lower ends of the fixed grate bars, carries the **clinker grinder,** the form of which varies with the service it is to perform. The clinker grinder is driven by a reciprocating bar running across the stoker front. This bar also controls the feed of the fuel and movement of the grate bars, and is driven by an electric motor or a steam engine located at one side of the setting.

A triple air-cooled arch, made in two parts and supported by the arch plate, extends over the whole grate. The lower part of the arch is made of firebrick and the upper arch of common brick. An air space, through which air is constantly circulating, separates these arches. Air is admitted to this air space through dampers on the front of the furnace, and then passes to the furnace through the passages formed by the ribs on

the arch plate. The front and sides of the furnace are covered with plates of sheet steel.

Operation. — Coal is delivered to the hopper either mechanically or by hand, and is intermittently fed, by the **pusher,** from the hopper to the coking plate at the upper end of the grate bars. While it is on the coking plate, the volatile gases are driven off by radiant heat from the white-hot arch, and are immediately mixed with the heated air admitted through the arch plate ducts, forming a combustible gas which is burned as it passes toward the rear of the furnace. The fuel travels slowly down the inclined grates toward the clinker grinder, receiving the necessary amount of air through the grate to complete the burning process. By the time it reaches the clinker grinder, combustion is complete, and the ash and refuse are automatically discharged into the ashpit.

The speed at which the pusher pushes the coal to the grates can be regulated as desired. The movable grate bars have their greatest action at their lower end, and thus break up the fuel bed for free admission of air to that part of the furnace where it is most required.

This stoker is suitable for all bituminous coals and gives satisfactory operation for uniform or variable loads. When properly operated, it produces little smoke. The depth of fire on the grates should be about 6 inches and the combustion rate about 23 pounds per square foot of grate surface per hour. The arch is essential in eliminating smoke. This stoker can be run at 150 to 200 per cent of rating.

180. Underfeed Stokers. — These are of two types: *one is a pure underfeed type,* such as the Jones, *in which the fuel bed is slightly rounded, and thicker at the center than on the sides; the other combines the underfeed*

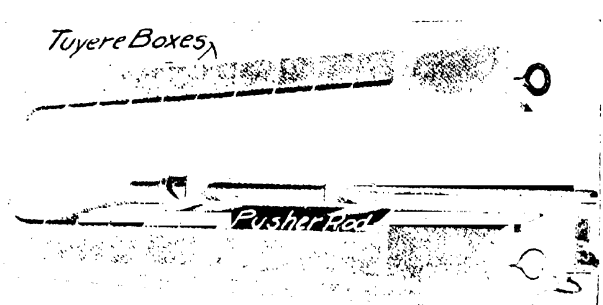


FIG. 106. — Inside Portion of Jones Underfeed Stoker.

principle with a gravity feed and has the fuel bed inclined from front to rear. The Riley stoker is typical of this class.

181. Jones Underfeed Stoker. — A Jones inclined stoker is shown in Fig. 2, page 6, installed under a Heine water-tube boiler. It consists of a retort

located inside the furnace, and an operating cylinder and attached parts outside the furnace.

The portion inside the furnace, Fig. 106, consists of a **heavy cast-iron retort,** or fuel magazine; hollow **cast-iron tuyere boxes** fastened to the side flange of the retort by lugs through which a retaining rod passes; an **auxiliary plunger rod** driven by the main ram, to which it is attached by an adjustable connection for changing the length of the stroke; a

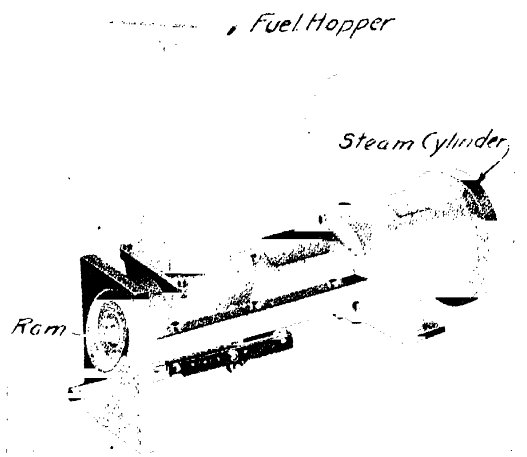


FIG. 107. — Outside Portion of Jones Underfeed Stoker.

pusher block bolted to the auxiliary plunger rod; and heavily **ribbed dead plates** which extend from the edge of the retort to the side wall of the furnace and support the ashes and refuse.

The portion outside the furnace, Fig. 107, consists of a **coal hopper;** a **ram case** directly below the hopper and bolted to the retort at the boiler front; a **ram** connected by a piston rod to a steam piston working in a cylinder attached to the ram case; and a **steam valve,** installed in the steam lines to the stoker cylinder, to control the speed of the piston driving the ram.

Operation. — The fuel bed, the appearance of which is shown in Figs. 108 and 109, consists of three zones, the green coal at the bottom, the coking zone in the middle, and the incandescent fuel bed at the top. Coal is fed into the hopper by hand or by mechanical means, and drops in front of the ram, which forces the coal into the retort when steam is admitted to the automatic control valves. The automatic control valve then reverses the plunger, and more coal falls into the ram case. The pusher blocks on the auxiliary plunger rod force the coal in the retort backward

and upward, thus breaking up the fuel bed at each stroke. Air is supplied by a fan at about 3 inches of water pressure, the exact pressure depending on the thickness of the fuel bed. The air enters below the stoker, passes to the tuyere boxes and into the fuel bed below the fire. Thus the volatile hydrocarbons from the coking zone are thoroughly mixed

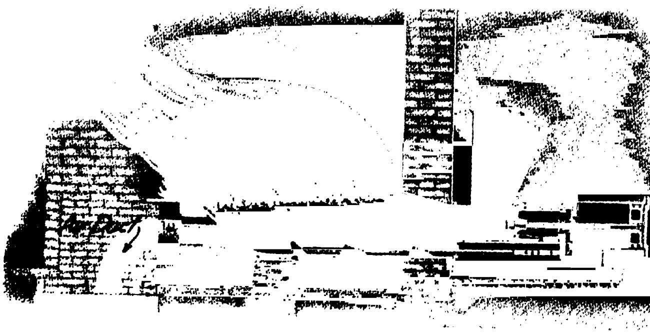


FIG. 108. — Longitudinal Vertical Section through Fuel Bed of Jones Stoker.

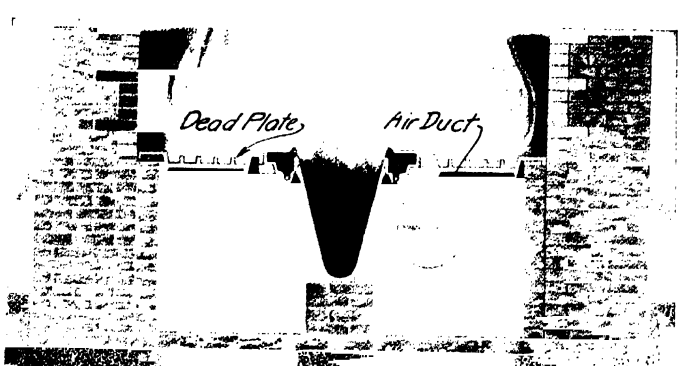


FIG. 109. — Vertical Cross-section of Jones Stoker.

with air before reaching the incandescent zone. The supply of air and fuel is automatically regulated by the steam pressure in the boiler, acting on control valves in the steam line to the stoker cylinder. The ashes collect on the dead plate and are removed through doors located at each side of the retort.

When larger capacity is desired, more than one retort is placed in the furnace, with a dead plate between each pair of retorts.

This type of stoker is adapted to bituminous coals, but not to anthracite. It does not require as large a combustion space as the overfeed stoker,

because the gases are burned in passing through the incandescent fuel bed. It can be applied in units from 75 horsepower, and upward. The thickness of the fire is about 15 inches.

In a recent development of the Jones stoker, the fuel bed and retort are given a slight incline from front to rear, with dump plates at the bottom of the incline. The ashes collect on dump plates, which are dumped by hand from the side of the furnace.

The latest development in this type of stoker is the **lateral-retort stoker,** which consists of one main retort extending from the front wall to the rear wall, with a number of lateral retorts at right angles to the main retort. Coal is fed into the main retort by a ram and plunger operated by a main steam cylinder. The coal is then fed sidewise into the lateral retorts by means of an auxiliary steam cylinder, which operates lateral pusher rods and blocks. Side dump plates are located at the end of the lateral retorts.

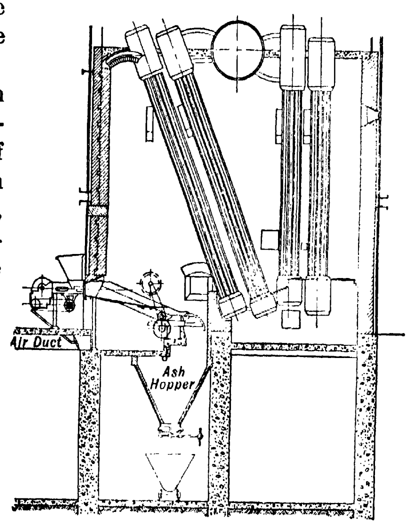


FIG. 110. — Riley Stoker under Biglow-Hornsby Boiler.

182. Riley Underfeed Stoker. — This stoker is shown in Fig. 110 under a Biglow-Hornsby water-tube boiler. It is a multiple-retort, inclined, underfeed stoker in which the coal is forced into the retorts of the furnace, Fig. 111, by a ram, from a point below that at which air is admitted.

That portion of the stoker which is outside the furnace is shown in Fig. 112. It consists of a **coal hopper** attached to the upper part of the **ram case**; **rams** driven by connecting rods from a crank shaft running the length of the stoker; an **enclosed speed-changing device** operated by a hand lever; a set of **change gears** connected to the crank shaft by a **worm** and **worm wheel**; a **change-gear shaft** operated by a steam engine or an electric motor; **handles** for changing stroke of underfeed grate blocks; **angle irons** running the length of the furnace to support the ram case front and rear; and **pipe supports** upon which the angle irons rest.

The portion of the stoker which is inside the furnace, Fig. 111, consists of **underfeed reciprocating grate bars,** Fig. 113, fastened to the retort

side bars by bolts; **side-bar shoes** to support the lower end of the side bar, **support rods** to carry the upper end and to give motion to the side bars; a **retort** formed between each two pairs of side bars and a metal

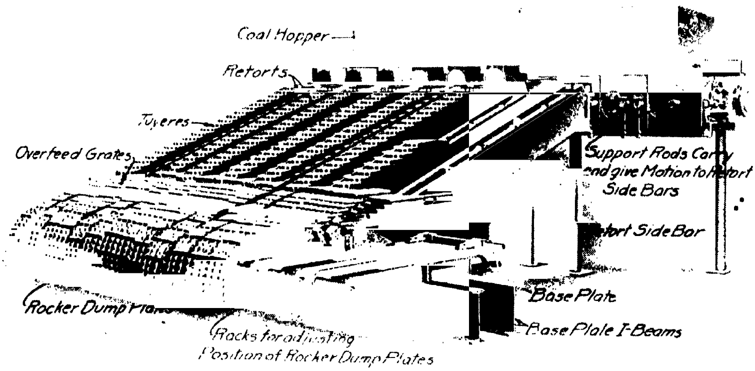


FIG. 111. — Perspective View of a 9-Retort Riley Underfeed Stoker.

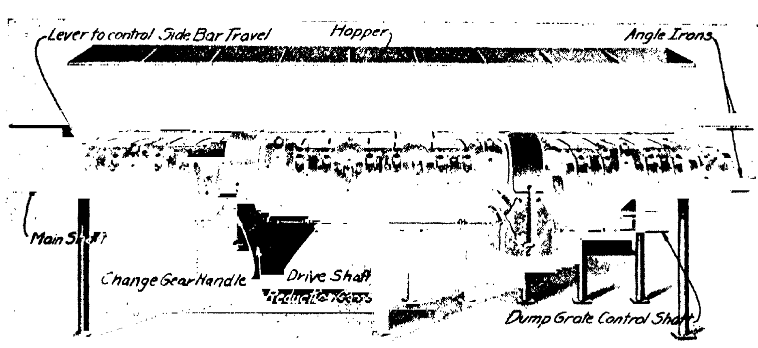


FIG. 112. — Riley Underfeed Stoker — Portion outside Furnace.

bottom to which are attached sifting strips to prevent sifting of coal into the space beneath; **overfeed reciprocating grates** attached to and moving with the side bars; **rocker dump plates** attached to the lower end of the side-bar shoes and adjustable by a rack connected to a shaft operated by a handwheel located at one side of the setting; a **base plate** upon which the side-bar shoe moves; and **two I-beams** upon which rest the base plate and dump plate racks.

Operation. — Coal is fed into the retorts from the hopper by rams, Fig. 114, which push the coal forward and upward. At the same time, the side bars are pushed forward, giving the coal a movement in the same direction. The overfeed reciprocating grates move the coal to the rocker dump plates, which continuously agitate, crush, and discharge the ash. The movement of the dump plates is adjustable, thus controlling the movement of the fuel bed and the dumping of refuse.

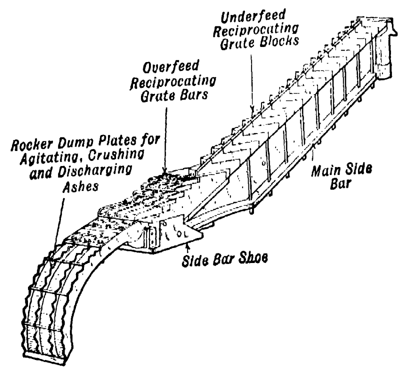


Fig. 113. — Reciprocating Grate Bar — Riley Stoker.

Distillation of the volatile gases occurs in the retorts, after which they are mixed with air discharged through openings, or tuyeres, under the grate blocks, which are carried by the moving side bars. The gases thus pass through an actual bed of burning coke and then through an incandescent fire zone.

Air for combustion is supplied, through air ducts, to the space below

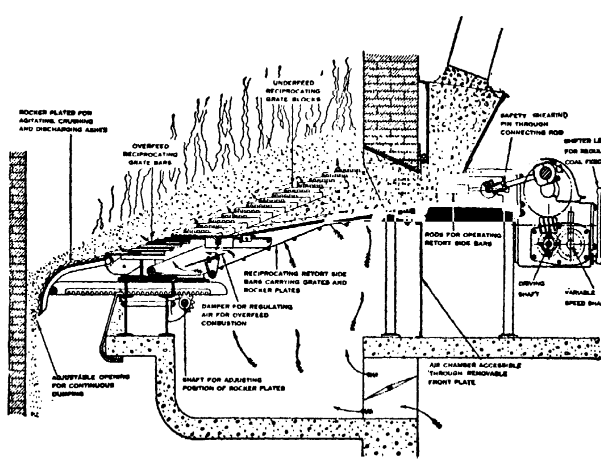


Fig. 114. — Section through Fuel Bed of Riley Stoker Fired Furnace.

the stoker. A steel plate covers the front of this space and prevents the escape of air. From this space the air then passes into the furnace, through

the tuyeres in the underfeed grates. The air to the overfeed grates is controlled by a damper.

This type of stoker is used mainly with boilers of large capacity, the size being determined by the number of retorts in the furnace. Each retort has a rating of about 100 horsepower, when not forced. This stoker is capable of forcing the boiler 300 to 350 per cent of its rating. Efficiencies as high as 80 per cent are obtainable when operating at normal loads; when forced, the efficiency is lowered slightly. All grades of free-burning bituminous coals can be used. The fuel bed varies from 2 to 3 feet in depth.

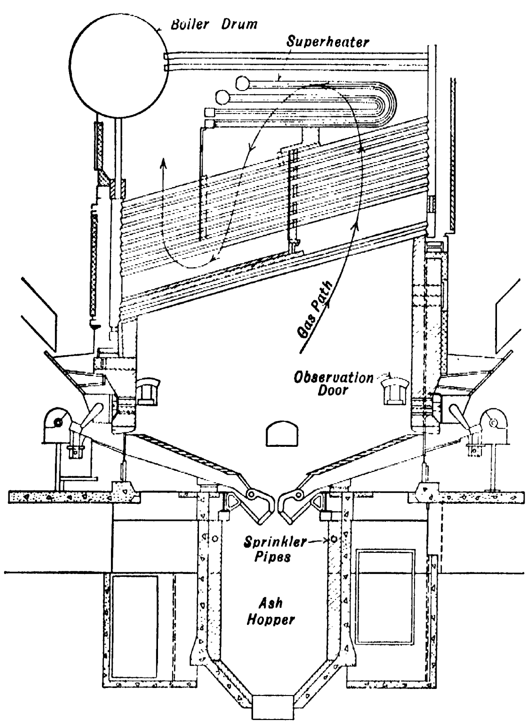


FIG. 115 — Westinghouse Duplex Underfeed Stoker.

The Taylor underfeed stoker was the pioneer stoker of the gravity underfeed type and it is used extensively in large Central Stations. It differs from the Riley stoker mainly in that the sides of the retorts remain stationary; a feeding ram and one or two distributing rams are located in each retort; the dump plates do not reciprocate but are dumped by either hand or power, and the rams are driven by a special form of connecting rod which has an attachment to each ram.

A duplex arrangement of a stoker of the gravity underfeed type, in which a bridge wall is not required, is shown in Fig. 115. This is a 24-retort Westinghouse stoker with power-operated dump grates, and is applied to a 1,140 horsepower cross-drum boiler at the Buffalo General Electric Co. It has an operating efficiency of 75 per cent at 200 per cent of rating and 65 per cent at 500 per cent of boiler rating. It is operating with coal having 10 per cent ash, 3 per cent moisture and 2 per cent sul-

phur. A Taylor duplex stoker installed at the Connor's Creek Station of the Detroit Edison Co. has 26 retorts and is applied to a 2365 horsepower Stirling boiler.

183. Stoker-fired Furnaces. — Furnaces equipped with stokers have been shown in Figs. 98, 99, 102, 110, and 115. The general form of the furnaces varies with the type of stoker. For the chain-grate and front-overfeed stoker, a coking arch is necessary to give satisfactory operation. The height and shape of the arch are varied to suit the fuel and the draft; with the chain grate it may be flat or inclined upward toward the rear, and only extends over a portion of the grate; with a side-overfeed stoker, an arch extending over the entire grate is necessary to prevent excessive formation of smoke. An underfeed stoker does not require an arch, as the smoke-making constituents are burned in passing through the incandescent fuel zone. All stoker furnaces require a larger volume than hand-fired furnaces, on account of the more rapid evolution of the volatile gases. The distance from fuel bed to boiler heating surface varies, and should be from 5 to 20 feet for satisfactory operation with water-tube boilers and bituminous coal. Recent installations have an unusually large furnace volume per square foot of heating surface; in a typical case this ratio is 0.42 to 1.

184. Stoker Comparisons. — Data compiled by R. J. S. Pigott, pertaining to the various types of stokers, are given in Table 18. The prices are those given in 1914.

TABLE 18. — STOKER DATA

Type of Stoker	Step and slope overfeed	V overfeed	Chain overfeed	Gravity underfeed	Horizontal retort underfeed
Average price per rated boiler horsepower	\$3.60	\$3.60	\$3.50–\$6.55	\$5.65	\$4.44
Normal forcing ability in per cent of rating	190	175	260	300–350	300
Price per maximum horsepower developable	\$1.90	\$2.06	\$2.52	\$1.62–\$1.88	\$1.48
Maintenance per ton coal fired, in cents	10–12	11–14	6–10	2.5–4	4–6
Attendance in man-hours per active hour	0.45	0.45–0.50	0.20–0.30	0.08–0.10	0.30–0.40
Pounds coal per square foot grate surface (maximum)	35–38	35–42	45–48	60–75	50–65

185. Locomotive Stokers. — Stokers are used on locomotives to lighten the labor of the firemen and to give a greater capacity than that attained by hand firing. There are a number of makes of locomotive stokers, differing mainly in the method used to feed the coal into the firebox.

A typical stoker, manufactured by the Locomotive Stoker Co., is shown in Fig. 116. It consists of:

1. A **conveyor** and **crushing system** for crushing and carrying the coal from the tender to the locomotive. The conveyor screw is operated by spur gears connected through a driving arm to a rack driven by the stoker engine. The speed of the conveyor shaft is regulated by the speed of the stoker engine, which is controlled by a hand-operated globe valve and operates at from 8 to 90 pounds per square inch pressure.

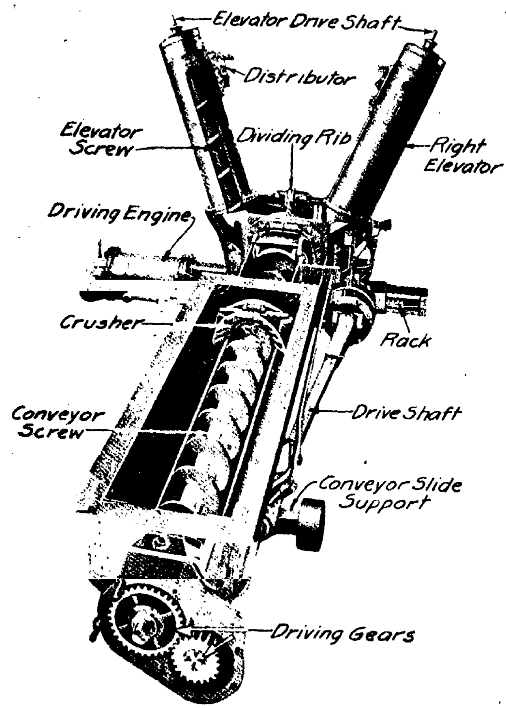


FIG. 116. — Rear View of Locomotive Duplex Stoker.

2. An **elevating system,** consisting of a transfer hopper located beneath the cab deck and containing an adjustable gate; and right and left elevator screws, driven by gears which mesh with a rack driven by the stoker engine, to raise the coal to the **distributors** set in the back boiler head.

3. A **distributing system,** which spreads the coal over the grate, using a low-pressure steam jet located in the back and bottom portion of the distributors. The grate is the same as those used for hand firing.

Operation. — The shovel sheet is provided with an opening, 18 inches wide, extending from the coal gates to the slope sheet of the feedwater tank. Coal passes through this opening to the trough beneath, and is carried by the conveyor screw through the crusher, where it is broken to a suitable size. Entering the transfer hopper, the coal is divided, equally or in the proportion desired, between the elevating screws, by a movable dividing rib. The elevating screws elevate the coal and drop it into tubes fitted with elbows which extend through the back head on each side of

the fire door. Steam jets in the elbows constantly blow the coal through the tubes to the distributors located inside the firebox. These distributors deflect and spread the coal over the entire surface of the fire. A level, light fire is carried, and excellent combustion results, the temperature being 400 deg. fahr. to 500 deg. fahr. higher than that attained with hand firing.

186. Methods of Burning Gas. — The burning of a gaseous fuel is accomplished by passing it into burners, where it is mixed with the required amount of air for combustion. The type of burner used depends upon the kind of gas to be burned.

Blast-furnace gas burners are of two general types, those in which the air for combustion is admitted around the burner proper, and those in which the air is admitted through the burner. Provision is made for regulating the air and gas supply independently. A gas opening of 0.8 square inch per rated horsepower will enable a boiler to develop its nominal rating with a gas pressure of 2 inches of water. The pressure is ordinarily about 6 to 8 inches of water, and thus the burner will be good for ordinary overloads. The size of the air openings varies from 0.75 to 0.85 of a square inch per rated horsepower. For good results the burners are slightly inclined downward toward the rear of the furnace.

For natural gas, a large number of small burners are used, each capable of handling 30 nominal rated horsepower. The use of a large number of burners obviates the danger of lancing or blow-pipe action, which would result if large burners were used. The gas pressure entering the burners is about 14 inches of water. The burners should give the gas and air a rotary motion to ensure proper mixture.

187. Furnaces for Gas Fuels. — The essential feature in gas-burning furnaces is sufficient combustion space to give the gases ample time to burn completely before striking the heating surfaces. For blast furnace gases, a furnace volume of 1 to 1.5 cubic feet per rated horsepower is satisfactory. For natural gas, a volume of 0.75 cubic feet gives good results. Gas furnaces ordinarily do not have a bridge wall.

188. Methods of Burning Oil Fuel. — Oil is burned under boilers by first atomizing or vaporizing it, so that it may be burned like gas. Burners of the following types are used to atomize the oil: (1) **Spray burners,** in which the oil is atomized by steam or compressed air; (2) **Mechanical burners,** in which the oil is atomized by submitting it to a high pressure and passing it through a small orifice.

The simplicity of the steam atomizer spray burner, the excellent economy of the better types, and the low oil pressure and temperature required, make it a favorite for stationary plants, where the loss of fresh water is not a vital consideration. In marine work, or in any case where it is necessary to save feedwater that otherwise would have to be added in

the form of "*make up*," either compressed air or mechanical means are used for atomizing. Compressed-air spray burners are in satisfactory operation in some stationary plants, but their use is not general.

Steam spray burners, as now used, may be divided into two classes: (1) Inside mixers in which the steam and oil come in contact within the

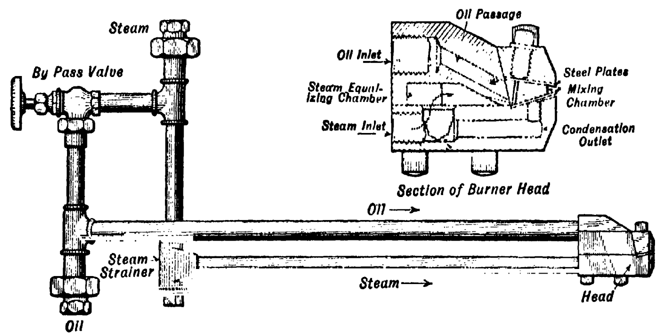


FIG. 117. — Hammel Steam Atomizing Oil Burner Complete.

burner and the mixture is atomized in passing through the orifice of the burner nozzle; (2) Outside mixers in which the steam flows through a narrow slot or horizontal row of small holes in the burner nozzle, the oil flowing through a similar opening above the steam orifice, and being picked up and atomized by the steam outside the burner.

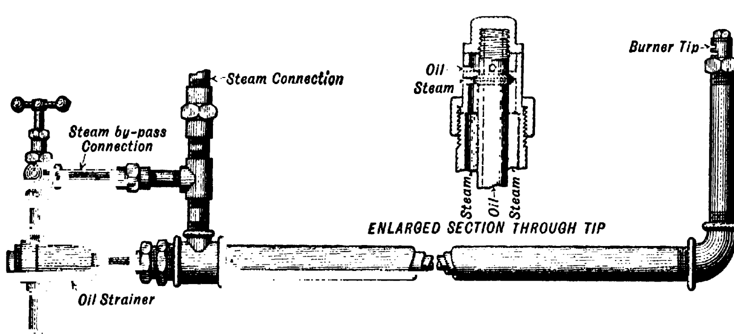


FIG. 118. — Peabody Steam Atomizing Oil Burner.

The Hammel oil burner, Fig. 117, belongs to the inside-mixing type. Steam, issuing from three small slots into the mixing chamber, cuts across the oil stream, and the energy of the steam is fully utilized in atomizing the heavy hydrocarbons and vaporizing the lighter ones. The mixture issues from the burner and ignites like a gas flame. The steam used

amounts to about 2 per cent of the steam generated by the boiler. This burner will handle heavy or light oils. The steam passage is continued into a condensation chamber which acts as a steam separator.

The **Peabody outside-mixing type** of oil burner, shown in Fig. 118, has given good satisfaction. The construction is clearly shown in the figure.

Mechanical burners, of which Fig. 119 is typical, have been used mainly on marine boilers. The most successful burners of this type have a round

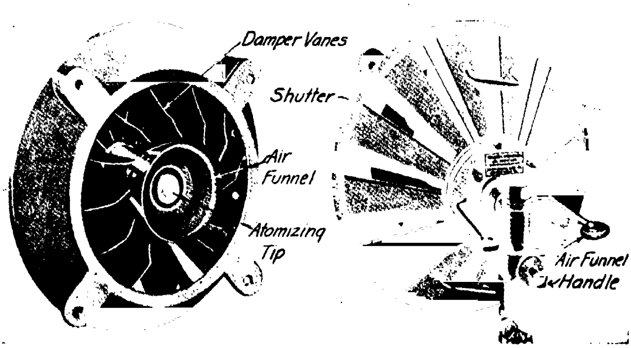


FIG. 119. — Mechanical Atomizing Fuel Oil Burner.

flame and give a whirling motion to the oil within the burner tip. The oil is delivered to the burner under a pressure that varies from 30 to 200 pounds per square inch and a temperature that varies from 150 deg. fahr. to 200 deg. fahr. depending upon the grade of oil and the make of the burner. A constant oil pressure is produced by a pump having a governor controlled by steam pressure. These burners are located at the front of the furnace, and the air required for combustion is admitted around the burner through suitable adjustable shutters of various shapes. This is a better means of regulation than that of the steam atomizer, where air is admitted through checkerwork. A forced blast of air is used to assist in giving a proper mixture of air and oil spray, and is especially necessary when operating at high ratings. Better regulation is possible with mechanical burners than with steam atomizing burners, because, where the former are installed, a larger number of burners are used per boiler horsepower.

189. Capacity of Oil Burners. — A good steam atomizing burner, properly located in a well-designed oil furnace, has a capacity of about 400 boiler horsepower. The nature of mechanical atomizing burners reduces the capacity of the individual burners to about 100 horsepower.

190. Oil Furnace. — *To burn oil successfully, the furnace must be properly proportioned. With a satisfactory burner the furnace arrangement and the method of introducing air for combustion are important factors.*

The point at which the atomized oil is introduced depends upon the type of boiler under which the burner is installed. With horizontal return-tubular, Fig. 120, and Heine boilers, the burners, when steam is used for atomizing, are located at the front of the furnace, with the flame directed toward the rear. No bridge wall is used, and the path of the gases and the volume of the furnace are satisfactory for complete combustion.

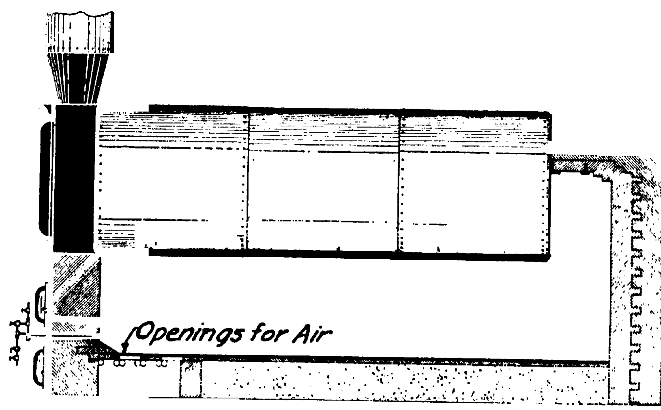


FIG. 120. — Hammel Oil Burner applied to H. R. T. Boiler.

For Stirling and horizontal drum water-tube boilers, this type of oil burner is located at the rear under the bridge wall, and the flame is directed toward the boiler front. The furnace thus increases in volume in the direction of the flame, and ensures free expansion and a thorough mixture of the oil and air. The gases have sufficient time to burn before coming into contact with the tube heating surfaces. If the flames are directed against the tubes they are soon burned out, because of the blowpipe action of the burner. The burner used in this type of furnace should have a form of flame that will not impinge directly on the heating surfaces. The burners should also be located so that the flames from the individual burners do not interfere or strike the side walls of the furnace. The burners are operated from the boiler front, and peep-holes are supplied through which the operator may watch the flame while he is adjusting it. Air for combustion is admitted through a checkerwork of firebrick supported on the furnace floor, the openings in the checkerwork being near the burner discharge.

With oil as a fuel, the efficiency of the boiler is generally high, because

combustion is more nearly perfect. Ordinarily, 14 per cent carbon dioxide can be attained, with only a trace of carbon monoxide.

191. Pulverized-fuel Burning. — The pulverized fuel is fed at a uniform rate from an overhead closed bunker to the burner or burners, by a motor-driven screw conveyor, or feeder. The amount of coal fed is regulated by means of hand-controlled side-dampers. From the end of the feeder the fuel falls through a pipe to the burner, Fig. 121, bolted to the front of the furnace.

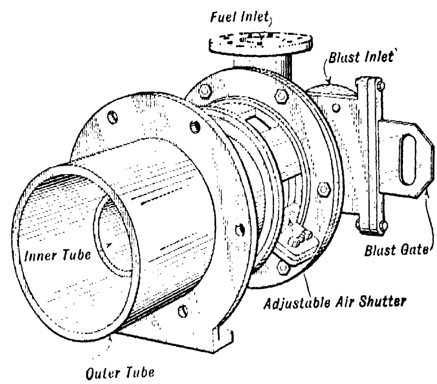


FIG. 121. — Pulverized Fuel Burner.

This burner has an inner and an outer tube. The inner tube is provided with a vertical opening for fuel and a horizontal opening connected to a blower. The blower supplies air to the burner at a pressure of about 5 inches of water, the amount of this air pressure being regulated by a **blast gate.**

A **deflector plate** at the bottom of the coal inlet ensures a thorough mixture of the pulverized coal with the air, under pressure, which acts as a carrying medium to project the fuel into the furnace. The outer tube has an adjustable register, to provide sufficient atmospheric air for combustion; the inner tube does not enter the furnace. The mixture of coal and air enters the furnace at the low pressure of $\frac{1}{2}$ inch of water. The blower supplies about 30 per cent of the required air, and the other 70 per cent is taken through the adjustable register or through damper-controlled air-tubes located at the front of the furnace.

The fuel, mixed with the proper amount of air for combustion, enters the combustion chamber in a finely divided state and is burned like a gas, while suspended in the air and before it comes into contact with the cooler surfaces of the boiler.

About 30 per cent of the ashes resulting from the combustion are carried from the chimney; the remaining 70 per cent are deposited in the soot chamber, on the tubes, or in the ashpit. The ashes which collect in the ashpit will fuse and produce slag in an improperly designed furnace. In a medium-sized plant the ashes should be removed once a day, and the tubes should be blown once in eight hours.

190. Oil Furnace. — *To burn oil successfully, the furnace must be properly proportioned. With a satisfactory burner the furnace arrangement and the method of introducing air for combustion are important factors.*

The point at which the atomized oil is introduced depends upon the type of boiler under which the burner is installed. With horizontal return-tubular, Fig. 120, and Heine boilers, the burners, when steam is used for atomizing, are located at the front of the furnace, with the flame directed toward the rear. No bridge wall is used, and the path of the gases and the volume of the furnace are satisfactory for complete combustion.

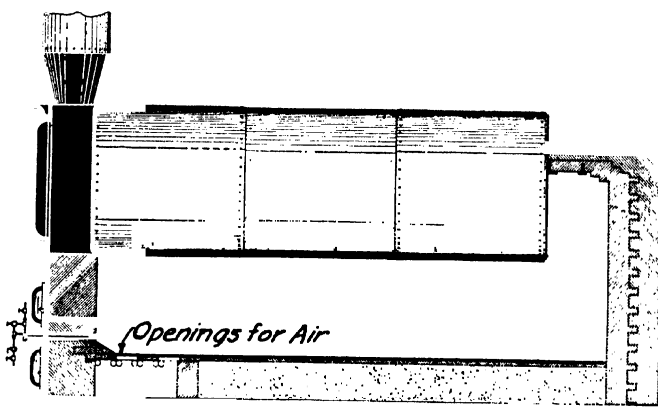


FIG. 120. — Hammel Oil Burner applied to H. R. T. Boiler.

For Stirling and horizontal drum water-tube boilers, this type of oil burner is located at the rear under the bridge wall, and the flame is directed toward the boiler front. The furnace thus increases in volume in the direction of the flame, and ensures free expansion and a thorough mixture of the oil and air. The gases have sufficient time to burn before coming into contact with the tube heating surfaces. If the flames are directed against the tubes they are soon burned out, because of the blowpipe action of the burner. The burner used in this type of furnace should have a form of flame that will not impinge directly on the heating surfaces. The burners should also be located so that the flames from the individual burners do not interfere or strike the side walls of the furnace. The burners are operated from the boiler front, and peep-holes are supplied through which the operator may watch the flame while he is adjusting it. Air for combustion is admitted through a checkerwork of firebrick supported on the furnace floor, the openings in the checkerwork being near the burner discharge.

With oil as a fuel, the efficiency of the boiler is generally high, because

required for the greater amount of fuel burned at the higher ratings. Vertical baffle walls are preferable with this type of furnace.

The cost of installation and operation compares favorably with those of an underfeed stoker. The overall efficiencies are slightly higher for the same capacity, and existing plants have attained an overall efficiency above 85 per cent. Such plants do not have standby losses caused by change of load or banked fires.

193. Methods of Smoke Determination. — No satisfactory method of determining the amount of smoke, either qualitatively or quantitatively, has yet come into use; and it cannot even be said that a reliable method of fixing the relative amount of smoke has been found. The condition of the atmosphere, the appearance of the background, and the personal equation all enter into the making of a determination. For qualitative smoke measurement, the Ringelmann chart, Fig. 123, is generally used.

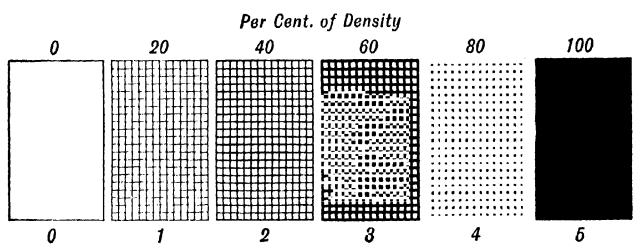


FIG. 123. — Ringelmann Chart for Grading the Density of Smoke.

This consists of four large charts ruled with vertical and horizontal lines forming squares, and numbered from 1 to 4, No. 1 representing a smoke density of 20 per cent; No. 2, 40 per cent; No. 3, 60 per cent; and No. 4, 80 per cent. A white card is 0 per cent density and a black card 100 per cent. These charts, together with the white and black cards, are placed 50 feet from the observer, in line with the chimney. The lines on the chart disappear at this distance and the charts have a gray appearance. The observer glances rapidly from the chimney to the charts and judges which chart corresponds, in color, with the smoke. He then records his observation and the time, on a smoke record chart. Readings are taken at one-half or one minute intervals.

In some cities, notably Chicago, a "**smoke unit**" has been adopted as a method of estimating the total smoke emitted by a chimney. This unit is the equivalent of No. 1 smoke emitted for one minute. Thus, if a stack emits No. 2 smoke for ten minutes it would be emitting 20 smoke units.

REFERENCES

Finding and Stopping Waste in Modern Boiler Rooms, H. S. BOILER WORKS.
Steam Power Plant Engineering, GEBHARDT.
Other references as under Chapter VI.

REVIEW QUESTIONS AND PROBLEMS

1. Explain the cause of smoke, and state the factors necessary for its prevention.

2. Name the types of grates used with hand-fired furnaces.

3. What three methods are used to fire coal by hand?

4. With hand-fired furnaces, how may the amount of smoke be minimized?

5. What are the advantages of mechanical methods of firing coal?

6. Classify mechanical stokers.

7. Describe the following types of stokers: (*a*) chain grate, (*b*) Riley underfeed, (*c*) Murphy.

8. Find the amount of coal burned per hour on a 5 ft. by 12 ft. chain grate which moves 4 in. per minute and has a depth of fuel bed of 8 in. Average weight of coal on grate, 45 lb. per cu. ft.

9. What can be said regarding the volume of stoker-fired furnaces when compared with hand-fired furnaces?

10. Describe a powdered-coal-burning furnace.

11. Describe an oil-burning furnace.

12. Explain the meaning of the "smoke unit," as used in Chicago.

CHAPTER IX

RATING AND EFFICIENCY OF STEAM BOILERS

194. Boiler Rating. — It is customary to rate steam boilers in the following ways:

1. By boiler horsepower
2. By amount of heating surface.
3. By area of grate surface.

195. Boiler Horsepower (*boiler hp.*). — The term **horsepower,** as applied to a boiler, designates the **capacity** of a boiler as regards its ability to evaporate a definite amount of water per hour, under a given set of conditions. *The term has no definite relation to the term horsepower as applied to a steam engine.* The boiler does not perform work, as ordinarily considered, but furnishes a medium that enables the steam engine to perform work.

At the Centennial Exposition in Philadelphia, in 1876, *a boiler horsepower was taken as the evaporation of* 30 *pounds of water into steam, at a gage pressure of* 70 *pounds per square inch with a feedwater temperature of* 100 *deg. fahr.* This was the weight of steam used, at that time, by a simple steam engine in producing one horsepower when the steam pressure was 70 pounds per square inch, gage. This rating is known as the **Centennial rating,** and gave a standard for comparing boiler performances.

The **commercial rating** of a boiler, at the present time, is the standard adopted by the American Society of Mechanical Engineers, and known as the A. S. M. E. standard.* According to this rating, a *boiler horsepower is the evaporation of* 34.5 *pounds of water into steam, from and at* 212 *deg. fahr. per hour.* Expressed differently, a boiler, to develop one boiler horsepower, must convert 34.5 pounds of water into steam per hour, when the feedwater temperature is 212 deg. fahr. and the absolute pressure 14.7 pounds per square inch; that is, the boiler must supply the latent heat of steam at atmospheric pressure for each pound of water evaporated.

The heat required to produce a boiler horsepower is, therefore, 34.5 × 971.7, or **33,523.7 B.t.u. per hour.** The horsepower rating of a boiler is found by multiplying the weight of water evaporated per hour by the

* The American Society of Electrical Engineers have proposed that the **myriawatt** be adopted as a standard for rating boilers. A myriawatt is the power equivalent of 10,000 watts, or 34,150 B.t.u. per hour.

heat required to evaporate a pound of the water and dividing this product by the heat corresponding to a boiler horsepower.

As steam may exist in a wet, dry, or superheated condition, the boiler horsepower may be found by using one of the following equations:

$$\text{Wet steam,} \qquad \textit{Boiler hp.} = \frac{W\,(xL + h - h_1)}{34.5 \times 971.7} \quad \text{(60)}$$

$$\text{Dry steam,} \qquad \textit{Boiler hp.} = \frac{W\,(H - h_1)}{34.5 \times 971.7} \quad \text{(61)}$$

$$\text{Superheated steam,} \; \textit{Boiler hp.} = \frac{W\,[H + C_p\,(t_s - t_v) - h_1]}{34.5 \times 971.7} \quad \text{(62)}$$

in which

W = total weight of water evaporated per hour, lb.
x = quality of the steam.
L = latent heat of steam at absolute boiler pressure.
h = heat of liquid corresponding to absolute boiler pressure.
h_1 = heat of liquid corresponding to feedwater temperature.

Note. — [$(t - 32)$ can be used for h_1 without serious error.]

H = total heat per pound of dry steam at absolute boiler pressure.
t_v = temperature of saturated steam at absolute boiler pressure, deg. fahr.
t_s = temperature of superheated steam at absolute boiler pressure, deg. fahr.
c_p = specific heat of superheated steam at constant pressure. (See Fig. 70, page 95.)

Example 24. — During the test of a 150 hp. horizontal return-tubular boiler, the following data were taken:

Water evaporated per hour, 3030 lb.; steam pressure, 110 lb. per sq. in. gage; barometer 29.50 in. mercury; moisture in steam 2 per cent; feedwater temperature 162 deg. fahr. Find the boiler horsepower developed.

Solution. — Using Equation (60), because the steam is wet,

$$\text{Boiler hp.} = \frac{W(xL + h - h_1)}{34.5 \times 971.7} = \frac{3030\,(0.98 \times 877.2 + 314.7 - 130)}{34.5 \times 971.7} = \mathbf{94.5}$$

W = 3030 lb.; x = 0.98
Absolute pressure in boiler, lb. per sq. in. = 29.50 × 0.491 + 110 = **124.5.**
L corresponding to 124.5 lb. per sq. in. = 877.2 B.t.u.
h corresponding to 124.5 lb. per sq. in. = 314.7 B.t.u.
h_1 corresponding to 162 deg. fahr. = $(t - 32)$ = 162 − 32 = 130 B.t.u.

196. Heating Surface, *H. S.* — *Manufacturers base their ratings of boiler horsepower on a definite amount of water-heating surface per boiler horsepower.* Formerly this amount of H. S. was 10 square feet for water-tube boilers, 12 square feet for tubular boilers, and 8 square feet for Scotch marine boilers. It is now customary to rate all stationary boilers on a

basis of 10 square feet of heating surface per boiler hp. This does not mean that the boiler should not develop more than its rating; but that, under ordinary operating conditions, 10 square feet of heating surface will evaporate the equivalent of 34.5 pounds of water into dry steam at 212 deg. fahr., when the feedwater temperature is 212 deg. fahr. This method of rating is arbitrary and is used for convenience only. It is known as the **manufacturer's rating** and is the **nominal rated capacity** of the boiler.

By this method, all boilers of equal heating surface are given the same rating. This is unfair to purchasers, as, of two boilers having equal heating surface, one may have its heating surface arranged more advantageously than the other and hence have greater capacity under the same conditions of operation.

The rated capacity may be exceeded by forcing the fire and improving furnace conditions. Engineers ordinarily select boilers on a basis of heating surface sufficient to supply the demand, allowing a reasonable rate of evaporation per square foot of heating surface per hour.

197. Heating Surface of H.R.T. Boiler. — *The heating surface of a return-tubular boiler equals one-half the external area of the cylindrical shell, plus the inside area of all the tubes, plus two-thirds the area of the rear head, minus the combined external cross-sectional area of the tubes, all expressed in square feet.*

Example 25. — Find the rated boiler horsepower of a boiler 60 inches diameter, 16 feet long, and having 70 three-inch tubes.

Solution. — The heating surface is found by applying the above rule.

1. Area of shell $= \frac{1}{2}\left(3.14 \times \frac{60}{12} \times 16\right) = 125$

2. *Area of all the tubes $= 3.14 \times \frac{2.78}{12} \times 16 \times 70 = 788$

3. Two-thirds area of rear head, less tube sections

$$= \tfrac{2}{3}\left[\tfrac{1}{4} \times 3.14 \times \left(\frac{60}{12}\right)^2\right] - \left[70 \times \tfrac{1}{4} \times 3.14 \times \left(\frac{3}{12}\right)^2\right] = 9.5$$

Total H. S. $= 922.5$

Allowing 10 square feet of heating surface per boiler horsepower, the rating equals approximately 92 boiler horsepower.

An approximate rule, which is sometimes sufficiently accurate to use in finding the total heating surface of this type of boiler, is to divide the area of the tubes by 0.85.

198. Heating Surface of a Water-tube Boiler. — For a water-tube boiler of the Heine type, *the heating surface equals one-half the area of the shell, the outside area of the tubes, the inner area of each water leg minus twice the area of all the tubes.*

An approximate rule is to divide the area of the tubes by 0.90.

* For data on tubes consult Table 19.

TABLE 19. — LAP-WELDED CHARCOAL-IRON BOILER TUBES

Diameter		Thickness, Inches	Wire Gage Number	Circumference		Transverse Areas			Length of Tube per Sq. Foot of	
External, Inches	Internal, Inches			External, Inches	Internal, Inches	External, Sq. Inches	Internal, Sq. Inches	Metal, Sq. Inches	Ex. Surf., Feet	Int. Surf., Feet
1	.856	.072	15	3.142	2.689	.785	.575	.21	3.819	4.462
$1\frac{1}{4}$	1.106	.072	15	3.927	3.475	1.227	.961	.266	3.056	3.453
$1\frac{1}{2}$	1.334	.083	14	4.172	4.191	1.767	1.398	.369	2.547	2.863
$1\frac{3}{4}$	1.56	.095	13	5.498	4.901	2.405	1.911	.494	2.183	2.448
2	1.81	.095	13	6.283	5.686	3.142	2.573	.569	1.909	2.11
$2\frac{1}{4}$	2.06	.095	13	7.069	6.472	3.976	3.333	.643	1.698	1.854
$2\frac{1}{2}$	2.282	.109	12	7.854	7.169	4.909	4.09	.819	1.528	1.674
$2\frac{3}{4}$	2.532	.109	12	8.639	7.954	5.94	5.035	.905	1.389	1.509
3	2.782	.109	12	9.425	8.74	7.069	6.079	.99	1.273	1.373
$3\frac{1}{4}$	3.01	.12	11	10.21	9.456	8.296	7.116	1.18	1.175	1.26
$3\frac{1}{2}$	3.26	.12	11	10.996	10.241	9.621	8.347	1.274	1.091	1.172
$3\frac{3}{4}$	3.51	.12	11	11.781	11.027	11.045	9.676	1.369	1.018	1.088
4	3.732	.134	10	12.566	11.724	12.566	10.939	1.627	.955	1.024
$4\frac{1}{2}$	4.232	.134	10	14.137	13.295	15.904	14.066	1.838	.849	.902
5	4.704	.148	9	15.708	14.778	19.635	17.379	2.250	.764	.812
6	5.67	.165	8	18.85	17.813	28.274	25.429	3.025	.637	.673
7	6.67	.165	8	21.991	20.954	38.485	34.942	3.543	.546	.573
8	7.67	.165	8	25.133	24.096	50.266	46.204	4.062	.477	.498
9	8.64	.18	7	28.274	27.143	63.617	58.629	4.988	.424	.442
10	9.594	.203	6	31.416	30.14	78.54	72.292	6.248	.382	.398

199. Grate Surface, *G. S.* — In rating boilers by grate surface, one-third of a square foot of grate surface is sometimes taken as equivalent to a boiler horsepower. This value varies, however, with the conditions under which a boiler is operated, and also differs with the method of firing. *The grate surface of a stoker-fired furnace is taken as the width between the furnace walls multiplied by the length of the furnace.* For inclined-grate stokers, the grate surface would thus be the projected area of the grate.

200. Ratio of Heating Surface to Grate Surface. — The ratio of heating surface to grate surface is a variable quantity. It varies with the type of boiler, and even among boilers of the same type. An average value for water-tube boilers is 40 to 1 and for fire-tube boilers, 50 to 1.

201. Equivalent Evaporation, W_e. — *The economy of a steam boiler is defined as the number of pounds of water evaporated per pound of coal used.* The heat required to evaporate one pound of water, from a feedwater temperature of 60 deg. fahr. into steam at 175 pounds per square inch absolute, is different from that necessary to evaporate one pound of water under other conditions; as for example, water at a temperature of 180 deg. fahr. into steam at a pressure of 150 pounds per square inch absolute. The number of pounds of water evaporated under the first condition represents a different quantity of heat from that required to evaporate the same number of pounds under the second condition. A comparison of boiler

performances under these two sets of conditions would be without value; to make a just comparison, the boiler performance must be reduced to some standard condition. The standard adopted is called **equivalent evaporation from and at 212 deg. fahr.** *By this standard the water actually evaporated is expressed in terms of the amount of water which would be evaporated, if the pressure were 14.7 pounds per square inch absolute and the feedwater temperature 212 deg. fahr., by using a quantity of heat equal to that used under the actual conditions of pressure and temperature.* Each pound of equivalent evaporation represents the addition of 971.7 B.t.u., or the latent heat of evaporation at a pressure of 14.7 pounds absolute.

The equivalent evaporation may be expressed thus:

Wet steam, $$W_e = \frac{W\,(xL + h - h_1)}{971.7} \qquad (63)$$

Dry Steam, $$W_e = \frac{W\,(H - h_1)}{971.1} \qquad (64)$$

Superheated steam, $$W_e = \frac{W\,[(H + C_p\,(t_s - t_v) - h_1)]}{971.7} \qquad (65)$$

in which W = actual weight of water fed per hour per pound of fuel, lb.
W_e = equivalent weight of evaporation per hour, lb.
971.7 = latent heat corresponding to a temperature of 212 deg. fahr., B.t.u.

Example 26. — Using data from Example 23, page 186, find the equivalent evaporation from and at 212 deg. fahr.

Solution. — Since the steam contains moisture, substitute the various values in Equation (63). The numerical values of all symbols are as in Example 23.

$$W_e = \frac{W\,(xL + h - h_1)}{971.7} = 3030 \times \frac{(0.98 \times 877.2 + 314.7 - 130)}{971.7} =$$
$$3030 \times 1.075 = 3257 \text{ lb. per hour.}$$

202. Factor of Evaporation, F. — *The factor of evaporation is the factor by which the actual evaporation is multiplied to obtain the equivalent evaporation.* It may be defined as the number of pounds of water that should be evaporated from a feedwater temperature of 212 deg. fahr. into dry steam at 212 deg. fahr., by the expenditure of the same amount of heat as was used to evaporate one pound of feedwater under the actual conditions. For the various states in which steam may exist, the factor of evaporation may be calculated from the following equations:

Wet Steam, $$F = \frac{xL + h - h_1}{971.7} \qquad (66)$$

Dry steam, $$F = \frac{H - h_1}{971.7} \qquad (67)$$

Superheated steam, $$F = \frac{H + C_p\,(t_s - t_v) - h_1}{971.7} \qquad (68)$$

Example 27. — The following data were taken during the test of a cross-drum B. and W. boiler, fired by Riley stokers, at the Columbus Railway, Power and Light Co.: Steam pressure, 256.5 lb. per sq. in. abs.; temperature of steam, 614 deg. fahr.; temperature of feedwater entering economizer, 128 deg. fahr. Find the factor of evaporation, including economizer and superheater.

Solution. — The steam is superheated, and Equation (68) will therefore be used.

$$F = \frac{H + C_p\,(t_s - t_v) - h_1}{971.7} = \frac{1200.8 + 0.56\,(614 - 403.3) - 95.9}{971.7} = 1.26$$

H at 256.5 lb. per sq. in. abs.	=	1200.8
t_v at 256.5 lb. per sq. in. abs.	=	403.3 deg. fahr.
t_s	=	614 deg. fahr.
C_p corresponding to temperature and pressure of superheated steam, from curve, page 95	=	0.56
h_1 at 128 deg. fahr.	=	95.9

203. Boiler Efficiency. — Efficiency is the ratio of the **output** to the **input.**

The **output of a boiler,** in heat units, is calculated from the weight of water evaporated. The calculation may be made in either of two ways: (1) *The equivalent evaporation,* W_e, *may be multiplied by* 971.7; or (2) *the actual evaporation,* W, *may be multiplied by the heat actually used in evaporating one pound of water under the existing conditions.* That is, the output may equal 971.7 W_e; or $W\,(xL + h - h_1)$ for wet steam, $W\,(H - h_1)$ for dry steam, and $W\,[H + C_p\,(t_s - t_v) - h_1]$ for superheated steam.

The **input to a boiler** may be based on the heat value of the fuel as fired; that is, the fuel that goes into the furnace door; or it may be based upon the heat evolved from the coal burned on the grate, not taking into account the heat value of the fuel which falls through the grate into the ashpit. The efficiency found by using the first value of input is the **overall efficiency of the boiler, furnace and grate,** and that found by using the second value is the **efficiency of the boiler and furnace,** excluding the grate. These efficiencies may be expressed as follows:

1. *Efficiency of boiler, furnace and grate*

$$= \frac{\textit{Output}}{\textit{Input}}$$

$$= \frac{\textit{Actual weight of water per hour} \times \textit{heat value per pound of steam}}{\textit{Weight of fuel per hour, as fired} \times \textit{heat value per pound of fuel, as fired}}$$

$$= \frac{W\,(xL + h - h_1)}{W_c C}, \text{ for wet steam} \quad (69)$$

2. *Efficiency of boiler and furnace*

$$= \frac{\textit{Actual weight of water per hour} \times \textit{heat value per pound of steam}}{\textit{Weight of combustible per hour} \times \textit{heat value per pound of combustible}}$$

$$= \frac{W\,(xL + h - h_1)}{W_c C - W_a C_a}, \text{ for wet steam} \quad (70)$$

in which W_c = weight of fuel per hour, lb.

C = heat value per pound of fuel, B.t.u.

W_a = weight of ash per hour, lb.

C_a = heat value per pound of ash. In practically all cases, the combustible in the ash is carbon, and C_a may be taken $= \dfrac{14{,}600 \times \text{per cent carbon in ash}}{100}$.

Other symbols are as in Art. **195**, page **186**.

In Equation (69), the weight of dry coal × heat value per pound of dry coal may be used for the input. The final result is, however, the same. The heat value per pound of steam should correspond to the condition of the steam; that is, wet, dry or superheated.

The year-round efficiency of boiler, furnace and grate does not ordinarily exceed 60 per cent.

Example 28. — Additional data from the boiler test of Example 27, page 190, are as follows: Weight of coal as fired, 5,460 lb. per hour; actual evaporation, 36,680 lb. per hour; heat value of coal as fired, 10,775 B.t.u.; weight of ashes, 965 lb. per hour; per cent carbon in ash, 20.26; remaining data as in Example 27. Find: (*a*) the overall efficiency of bo ler plant, and (*b*) efficiency of boiler and furnace.

Solution. — (*a*) Using Equation 69.

Efficiency, boiler, furnace and grate

$$= \frac{W\,[H + C_p\,(t_s - t_v) - h_1]}{W_c C} = \frac{36{,}680 \times 1222.8}{5460 \times 10{,}775} = 0.764$$

$W = 36{,}680$ lb. per hr.; $W_c = 5460$ lb.; $C = 10{,}775$ B.t.u.

$[H + C_p\,(t_s - t_v) - h_1]$ from *Example* 27, $= 1222.8$.

(*b*) Efficiency, boiler and furnace

$$= \frac{W\,[H + C_p\,(t_s - t_v) - h_1]}{W_c C - W_a C_a} = \frac{36{,}680 \times 1222.8}{5460 \times 10{,}775 - 965 \times 2964} = 0.803$$

All values as in (*a*), except $W_a = 965$ lb. per hour, and

$$C_a = \frac{14{,}600 \times 20.3}{100} = 2964 \text{ B.t.u.}$$

204. Effect of Capacity on Efficiency. — When a boiler is forced beyond its normal rated capacity, its efficiency is usually lowered. The decrease in efficiency will not be great until the boiler is 50 per cent or more overloaded. This drop in efficiency is caused by the inability of the boiler to absorb the excess heat produced when forcing the fires.

REFERENCES

Mechanical Equipment of Buildings, Vol. II, Harding and Willard.

Steam, Babcock and Wilcox Company.

REVIEW QUESTIONS AND PROBLEMS

1. What is the significance of the term "boiler horsepower"?

2. Define the A. S. M. E. standard for boiler horsepower. What is the relation it bears to the Centennial rating?

3. What is the standard amount of heating surface equivalent to one boiler horsepower?

4. The following data apply to Dillon return-tubular boilers:

	1	2	3	4	5
Length of tubes, ft.	16	15	19	17	22
Length of boiler (overall)	17	16′ 2″	20′ 2″	18′ 8″	23′ 10″
Diam. of shell, in.	36	54	60	84	96
No. of tubes	32	62	82	150	262
Size of tubes, in.	$2\frac{1}{2}$	3	3	$3\frac{1}{2}$	3

Find: (*a*) Square feet of heating surface in each boiler, and check with the approximate rule, Art. 197.

(*b*) Rated horsepower of each boiler.

5. What should be the grate area for each of the boilers in Problem 4?

6. The following data were obtained during the test of a Wickes vertical water-tube boiler: absolute steam pressure, 212.1 lb. per sq. in.; superheat, 162.2 deg. fahr.; temperature of feedwater, 198.8 deg. fahr.; water evaporated, 70,938 lb.; coal fired, 9,193 lb.; total ash and refuse, 953 lb.; per cent conbustible in ash, 8.68; heat value per pound of dry coal, 13,245 B.t.u; moisture in coal fired, 4.42 per cent; duration of test, ten hours. Find the following:

(*a*) Factor of evaporation.

(*b*) Total equivalent evaporation, including superheater.

(*c*) Overall efficiency of plant.

(*d*) Efficiency of boiler and furnace combined.

7. During the test of a B. and W. cross-drum marine boiler used on the U. S. S. Battleship Wyoming, the following data were taken: steam pressure, 209.9 lb. per sq. in. gage; feedwater temperature, 168.6 deg. fahr.; barometer, 30.0 in. mercury; weight of oil used, 5,943 lb.; quality of steam, 99.19 per cent; weight of water evaporated, 74,898 lb.; duration of test, two hours; heat value per pound of oil, 19,291 B.t.u.

Find:

(*a*) Equivalent evaporation from and at 212 deg. fahr.

(*b*) Factor of evaporation.

(*c*) Boiler horsepower developed.

(*d*) Overall efficiency.

8. During the test of a 468 horsepower boiler using pulverized coal, data were taken as follows: pressure in boiler, 167 lb. per sq. in. gage; barometer, 29.25 in. mercury; temperature of feedwater, 157.2 deg. fahr.; total weight of fuel, 47,775 lb.; total weight of water, 393,168; degrees superheat, 374.9 deg. fahr.; B.t.u. per pound fuel as fired, 10,779; duration of test, twenty-four hours. Find the items called for in Problem 7.

CHAPTER X

STEAM BOILER TESTING

205. Foreword. — There are many reasons for testing boilers, the most important of which, as given by the POWER TEST COMMITTEE of the A. S. M. E., are:*

1. To determine the capacity and efficiency of the boiler, for comparison with standard or guaranteed results.
2. To determine the cause of superior or inferior results.
3. To compare different kinds of fuels.
4. To compare different conditions and methods of operation.
5. To determine the effect of changes in design or proportion of the boiler or furnace, upon capacity and efficiency.

A boiler test generally consists of the making of certain observations, which are described in the present chapter. The person in charge of the test should have the aid of a sufficient number of assistants, so that he may be free to give his attention to any part of the work whenever and wherever it may be required. *He should make sure that the instruments and testing apparatus continually give reliable indications and that the readings are being correctly recorded.* He should also keep a close watch on the operation of the boiler under test, and see that the operating conditions determined upon are maintained, and that nothing occurs to vitiate the data.

The object of the test should be clearly kept in view at all times. *Accuracy and reliability must characterize the work from beginning to end.*

206. Preparation for the Test. — The dimensions of the principal parts of the apparatus which bear on the object in view should be measured, or determined from correct working drawings. The general features of the boiler should be observed, and sketches should be made to show unusual points of design.

The physical condition of all parts of the boiler and apparatus which concern the object in view should be thoroughly examined, and a record made of the conditions found, together with any points in the operation which would affect the results. The tubes and joints of the boiler should be examined for leakage, and the condition of brick furnaces, grates, and baffles noted. The brick walls and cleaning doors should be examined

* A considerable portion of the material in this chapter has been taken from the A. S. M. E. "Rules for Conducting Performance Tests of Power Plant Apparatus."

for air leaks, either by shutting the damper and observing the escaping smoke, or by the candle-flame test; the latter test is based upon the fact that the flame of a candle, if held near an opening into which air is leaking, will be drawn strongly toward the opening and thus indicate the point at which leakage is taking place.

The condition of heating surfaces, with reference to exterior deposits of soot and interior deposits of mud and scale, should be determined. The steam main should be so arranged that condensed and entrained water cannot flow back into the boiler. Whenever the highest efficiency and capacity is desired, any defects tending to make the results unfavorable should be repaired before starting the test. Leakage of steam through blow-offs, drips, or any steam or water connections of the apparatus undergoing test, which would in any way affect the results, should be prevented by blanking off; or satisfactory assurance should be obtained that there is no leakage, either outward or inward. *This is a most important matter, and no assurance should be considered satisfactory which is not susceptible of absolute demonstration.*

Before the start of the test, the boiler should have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout. For tests to determine efficiency, it is desirable to run preliminary tests to determine the most advantageous conditions of operation.

207. Character of Fuel. — For tests of maximum efficiency or capacity of the boiler, to be compared with those of other boilers, the coal should be of some kind which is commercially regarded as a standard for the locality where the test is made. In the **Eastern states** the standards thus regarded for semi-bituminous coals are Pocahontas (Va. and W. Va.) and New River (W. Va.); for anthracite coals those of the No. 1 buckwheat size, fresh-mined, containing not over 13 per cent ash by analysis; and for bituminous coals, Youghiogheny and Pittsburgh coals. There are no special grades of coal mined in the **Western states** which are widely and generally considered as standards for testing purposes, the best coal obtainable in any particular locality being regarded as the standard of comparison.

For guarantee and other tests in which a coal containing not more than a certain amount of ash and moisture is specified, the coal selected for the test should not be higher in ash and moisture than the stated amounts, because any increase is liable to reduce the capacity to an extent proportionally greater than the increase in the amount of ash and moisture.

The size of anthracite coal should be determined by screening a suitable sample.

Tests made with oil or gas fuels accord with the rules for making a test with coal. The "**running**" **method** of starting and stopping is used, and the length of the test may be shortened.

208. Apparatus and Instruments. — The weight of coal and ashes used on a test is generally obtained by using **platform scales,** which should be tested for accuracy by calibrating with standard weights.

The quantity of feedwater should be obtained by means of tanks and platform scales. A suitable arrangement for this purpose consists of one or more tanks, each resting on a platform scale, and elevated a sufficient distance above the floor to empty into a receiving tank below. The feed pump is connected to the lower tank. Each tank should be of sufficient size to give ample time for the proper weighing of the water between successive emptyings.

The feedwater is often measured by some form of **water-meter,** of which there are many commercial types. *The meter should be calibrated in place before and after test to ensure accuracy.*

One typical water-meter uses a **triangular weir** over which the water flows. The weight of water is then determined by calculation based on the height of the water surface above the bottom of the weir, as explained in Art. 441, page 440.

For measuring water flowing in pipes, the **Venturi meter,** Fig. 124, is well suited. This meter consists of a short constricted passage, or throat,

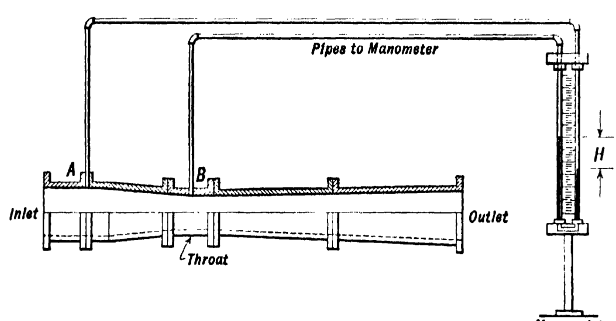


Fig. 124. — Venturi Meter.

joined to the full section of the pipe by conical converging or diverging sections. *The amount of water flowing is proportional to the difference in* **static,** *or radial, pressure at the sections indicated,* this pressure difference being obtained by a manometer. The meter is calibrated by weighing, and the amount of water flowing is found as follows:

$$Q = A_t V \qquad (71)$$

in which Q = cubic feet of water discharged per second.

A_t = area of throat section, square feet.

V = velocity of water at throat, feet per second.

$$= \frac{A_a}{\sqrt{A_a^2 - A_t^2}} \sqrt{2gH}$$

in which A_a = area of upstream section, square feet.

H = difference in pressure, measured in feet of water by the manometer.

The various pressures and temperatures are measured by pressure gages, draft gages, thermometers and pyrometers. These instruments are described in Chapter III.

The amount of moisture in the steam, when wet, should be determined by a throttling or separating calorimeter. When the steam is superheated, its temperature should be determined by a thermometer inserted in a thermometer well attached to the superheating surface.

The analysis of the flue gases is generally determined by some form of Orsat apparatus, Art. 158, page 139. If momentary samples are obtained, the analysis should be made as frequently as possible, say every fifteen or thirty minutes, and the conditions of the furnace and firing should be noted when the sample is withdrawn.

The amount of air used for combustion is often measured by an **anemometer,** Fig. 125, as the air enters the space beneath the grate. This instrument consists of a light vane which is revolved by the movement of the air, the number of revolutions being recorded by a system of gears and pointers. *The instrument measures the linear velocity only.*

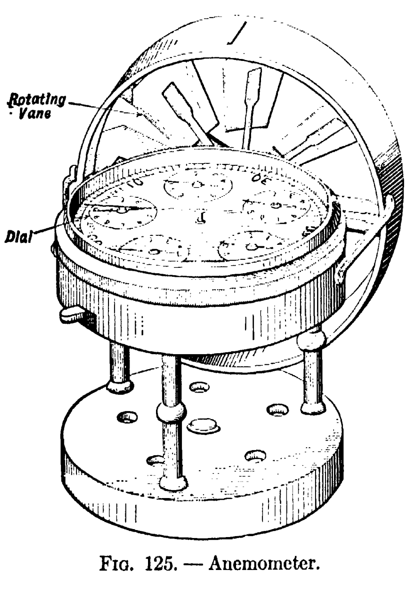


FIG. 125. — Anemometer.

All apparatus and instruments used in taking observations should be carefully calibrated, both before and after test, to ensure their accuracy.

209. Length of Test. — The duration of tests to determine the efficiency of a hand-fired boiler should be ten hours of continuous running, or such time as may be required to burn a total of 250 pounds of coal per square foot of grate.

In the case of a boiler using a mechanical stoker, the duration, where practicable, should be at least twenty-four hours. If the stoker is of a type that permits the quantity of coal and condition of the fuel bed at beginning and end of the test to be accurately estimated, the

duration may be reduced to ten hours, or such time as may be required to burn the above noted total of 250 pounds per square foot of grate surface.

In commercial tests where the service requires continuous operation night and day, with frequent shifts of firemen, the duration of the test, whether the boilers are hand-fired or stoker-fired, should be at least twenty-four hours. Likewise, in commercial tests, either of a single boiler or of a plant of several boilers, where the plant operates regularly a certain number of hours and during the rest of the day the fires are banked, the duration should not be less than twenty-four hours. The duration of tests to determine the maximum evaporative capacity of a boiler, without determining the efficiency, should not be less than three hours.

210. Starting and Stopping. — *The conditions, regarding the temperature of the furnace and boiler, the quantity and quality of the live coal and ash on the grates, the water level, and the steam pressure, at the beginning and at the end of the test, should be as nearly alike as possible.*

To secure the desired equality of conditions with hand-fired boilers, the following method should be employed: The furnace being well heated by a preliminary run, burn the fire low, and thoroughly clean it, leaving enough live coal, spread evenly over the grate, say 2 to 4 inches, to serve as a foundation for the new fire. Note quickly the thickness of the coal bed, as nearly as it can be estimated or measured; also the water level, the steam pressure, and the time, and record the latter as the starting time. Fresh coal should then be fired, from that weighed for the test, the ashpit thoroughly cleaned, and the regular work of the test put into operation. Before the end of the test, the fire should again be burned low and cleaned, in such a manner as to leave approximately the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly the water level, the steam pressure, and the time, and record the latter as the stopping time. *If the water level is not the same as at the beginning, a correction should be made by computation, rather than by feeding additional water after the final readings are taken.* Finally, remove the ashes and refuse from the ashpit and record their weight.

To obtain the desired equality in the condition of the fire, when a mechanical stoker, other than a chain grate, is used, the procedure should be modified, where practicable, as follows:

Regulate the coal feed so as to burn the fire to the low condition required for cleaning. Shut off the coal-feeding mechanism and fill the hoppers level full. Clean the ash or dump plate, note quickly the depth and condition of the coal on the grate, the water level, the steam pressure, and the time, and record the latter as the starting time. Then start the coal-feeding mechanism, clean the ashpit, and proceed with the regular work of the test. When the time arrives for the close of the test, shut off the coal-feeding mechanism, fill the hoppers, and burn the fire to the same

low point as at the beginning. When this condition is reached, note the water level, the steam pressure, and the time, and record the latter as the stopping time. Finally, clean the ash plate and haul the ashes. In the case of chain-grate stokers, the desired operating conditions should be maintained for half an hour before starting a test and for a like period before its close, the height of the throat plate and the speed of the grate being the same during both of these periods.

211. Operating Conditions. — In all tests in which the object is to determine the performance under conditions of maximum efficiency, or where it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit. In a stationary steam plant, for example, where maximum efficiency is the object in view, there should be uniformity in such matters as steam pressure, times of firing, quantity of coal supplied at each firing, thickness of fire, and in other firing operations; also in the rate of supplying the feedwater, in the load on the engine or turbine, and in the operating conditions throughout. On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity is either desired or required, unless this uniformity corresponds to the regular practice; when this is the object, the usual working conditions should prevail throughout the trial.

212. Records. — *A log of the data should be entered in notebooks or on blank sheets suitably prepared in advance.* This should be done in such manner that the test may be divided into hourly periods, or, if necessary, periods of less duration; the leading data may then be obtained for any period or periods, as desired, thereby showing the degree of uniformity obtained. A sample feedwater log is shown in Table 20.

The readings of the various instruments and apparatus concerned in the test, other than those showing quantities of consumption (such as fuel, water, and gas), should be taken at intervals not exceeding half an hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced, according to the extent of the fluctuation. In the case of smoke observations, for example, it is often necessary to take observations every minute, or still oftener, continuing these throughout the period covering the range of variations. *When it is essential that a number of instruments be read simultaneously, there should be an observer stationed at each one, and the readings should be taken at a signal from a time-keeper.*

Coal should be weighed and delivered to the firemen in portions sufficient for one hour's run, in order that the degree of uniformity of firing may be ascertained. An ample supply of coal should be maintained at

all times, but the quantity on the floor at the end of each hour should be as small as practicable, so that it may be readily estimated and deducted from the total weight, to determine the hourly weight.

The records should be such that the amount of feedwater used during each hour can be ascertained, thus giving a check on the uniformity of evaporation.

TABLE 20. — FEEDWATER LOG, BOILER TRIAL NO. 1

Made at............
Date...............
Boiler No.........

By..................
Fireman.............

Time	Water delivered to feed tank pounds	Temp. of water in tank ° F.	Time	Water delivered to feed tank pounds	Temp. of water in tank ° F.	Remarks
7:00	7855	196				Test began at 7.00 A. M.; March 26. Test closed at 6.00 P. M.; March 26.
8:30	3169	197				
9:00	3244	196				
9:40	2567	196				
10:00	5845	196				
11:00	5638	198				
12:01	6339	199				
1:00	6449	197				
2:03	9076	198				
3:45	3312	200				
4:03	3275	200				
4:52	2600	195				
5:04	4290	198				

Make a memorandum of every event connected with the progress of a test, however unnecessary it may appear at the time. A careful record should be made of the time of every such occurrence and the time of taking every weight and every observation. *For the purpose of identification, the signature of the observer, and the date, should be affixed to each log sheet or record.*

In the matter of weighing coal by the barrow-load, or weighing water by the tankful, which is required in many tests, a series of marks, or tallies, should never be trusted. The time each load is weighed or emptied should be recorded. *The weighing of coal should not be delegated to unreliable assistants, and, whenever practicable, one or more men should be assigned solely to this work. The same may be said with regard to the weighing of feedwater.*

213. Sampling and Drying Coal. — During the progress of the test, the coal should be regularly sampled for the purpose of analysis and determination of moisture.

Select a representative shovelful from each barrow-load, as it is drawn from the coal pile or other source of supply, and store the samples in a cool place in a covered metal receptacle. When all the coal has thus been sampled, break up the lumps, thoroughly mix the whole quantity, and finally reduce it, by the process of repeated quartering and crushing, to a sample weighing about 5 pounds, the largest pieces being about the size of a piece of pea coal. From this sample, two 1-quart, air-tight, glass fruit-jars, or other air-tight vessels, are to be promptly filled and preserved for subsequent determinations of moisture, calorific value, and chemical composition. These operations should be conducted where the air is cool and free from drafts.

When the sample lot of coal has been reduced by quartering to, say, 100 pounds, a portion weighing 15 to 20 pounds should be withdrawn for the purpose of immediate moisture determination.

214. Ashes and Refuse. — The ashes and refuse, withdrawn from the furnace and ashpit during the progress of the test and at its close, should

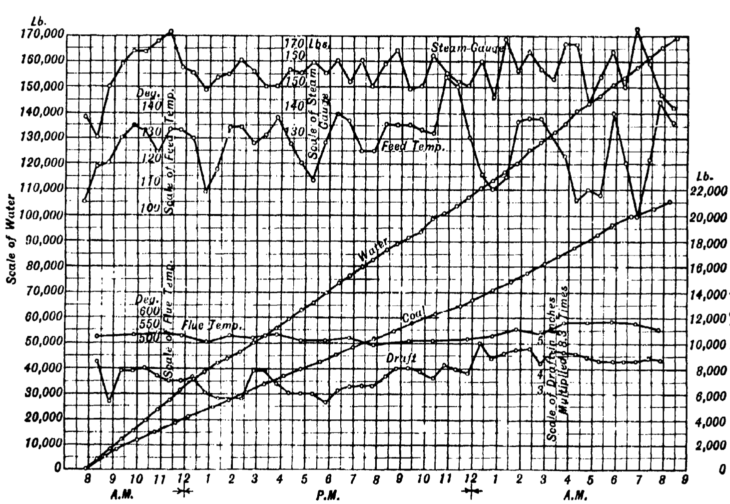


FIG. 126. — Boiler Data Chart.

be weighed, if possible, in a dry state. If this material is wet, the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may serve also for analysis and the determination of unburned carbon and fusing temperature.

Clinker should be weighed separately from the fine ash. The sample may be prepared as explained for coal, and should contain approximately the same proportions by weight of clinker and ash as for the whole test.

215. Smoke Observations. — In tests of bituminous coals, requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of five minutes, or, if necessary, every minute. At the same time, the furnace and firing conditions should be noted.

216. Data and Results. — It is well to plot all the data on a chart, Fig. 126, which shows the degree of uniformity of operation at a glance. This should be done during the progress of the test.

217. Calculation of Results. — The necessary calculations are made according to the principles explained in the previous chapters.

218. Report of Test. — A report of the test should present all the leading facts bearing on the design, condition, and operation of the apparatus tested, together with such sketches and photographs as may be needed for a clear understanding of all points of the test. It should state the object and character of the test, the methods followed, the conditions maintained, and the conclusions reached, and should close with a tabular summary of the principal data and results. The form of the data-and-result sheet is given in Table 21, in which two tests are recorded. The first test is for a boiler which has no superheater and no economizer; the second is for a boiler having both. Observed data are recorded in roman type, and calculated results in **bold-face type.**

Select a representative shovelful from each barrow-load, as it is drawn from the coal pile or other source of supply, and store the samples in a cool place in a covered metal receptacle. When all the coal has thus been sampled, break up the lumps, thoroughly mix the whole quantity, and finally reduce it, by the process of repeated quartering and crushing, to a sample weighing about 5 pounds, the largest pieces being about the size of a piece of pea coal. From this sample, two 1-quart, air-tight, glass fruit-jars, or other air-tight vessels, are to be promptly filled and preserved for subsequent determinations of moisture, calorific value, and chemical composition. These operations should be conducted where the air is cool and free from drafts.

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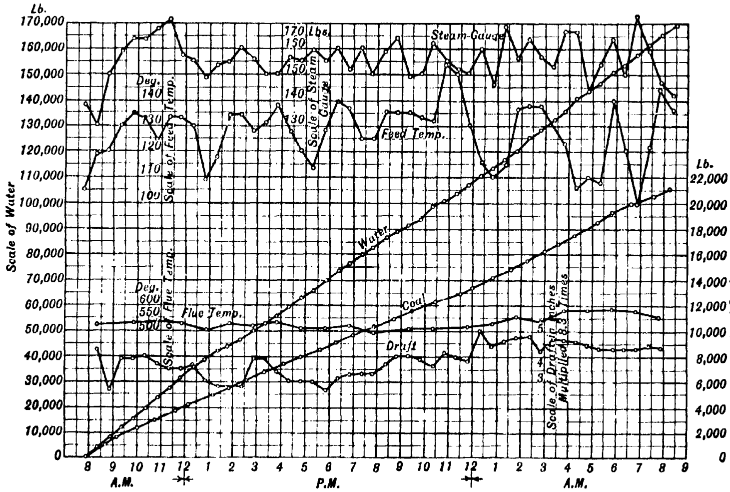

FIG. 126. — Boiler Data Chart.

be weighed, if possible, in a dry state. If this material is wet, the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may serve also for analysis and the determination of unburned carbon and fusing temperature.

Clinker should be weighed separately from the fine ash. The sample may be prepared as explained for coal, and should contain approximately the same proportions by weight of clinker and ash as for the whole test.

TABLE 21. — DATA AND RESULTS OF A BOILER TEST. (*Continued*)

Item	Name of Item with Units	1	2
20	Weight of coal as fired, lb.	8936	139,910
21	Weight of dry coal, lb.	8830	128,500
22	Weight of ash and refuse, lb.	840	28,350
23	Combustible in ash, per cent	.....	22.05
24	Weight of combustible burned, lb.	7990	100,150
25	Weight of dry coal consumed per hour, lb	1104	5355
26	Dry coal per sq. ft. of grate surface per hour, lb.	24.53	....
27	Combustible burned per hour, lb	.....	.....
28	Combustible burned per sq. ft. grate surface per hr., lb.	.....	.....
29	Analysis of flue gases, per cent by volume		
	Carbon dioxide	.....	12.10
	Oxygen	.....	6.93
	Carbon monoxide	.....	0.19
	Nitrogen (by difference)	.....	80.78
	Water, total for test		
30	Total weight of water fed to boiler, lb	86,872	907,500
31	Total weight of water evaporated, corrected for moisture, lb.	85,786	.....
32	Factor of evaporation	1.14	1.238
33	Equivalent water evaporated into dry steam from and at 212 deg. fahr., lb.	97,696	1,123,000
	Water, per hour		
34	Water evaporated per hour corrected for moisture, lb	10,723	37,830
35	Equivalent evaporation into dry steam per hour from and at 212 deg. fahr., lb.	12,212	46,800
36	Equivalent evaporation per hour from and at 212 deg. fahr. per sq. ft. of water heating surface, lb	5.98	10.63
37	Horsepower developed, hp.	356.8	1358
38	Builder's rated horsepower on basis of 10 sq. ft. per hour, hp.	204.0	444
39	Percentage of builder's rated horsepower developed	174.6	305
	Economic results		
40	Water apparently evaporated under actual conditions per pound of coal as fired, lb	9.74	6.49
41	Equivalent evaporation from and at 212 deg. fahr. per pound of coal as fired, lb.	10.90	8.05
42	Equivalent evaporation from and at 212 deg fahr. per pound of dry coal, lb	11.06	8.75
43	Equivalent evaporation from and at 212 deg. fahr. per pound of combustible, lb.	12.22	11.20
	Efficiency results		
44	Efficiency of boiler, furnace and grate, per cent	72.8	72.2
45	Efficiency of boiler and furnace, per cent	76.4	79.2

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Power Plant Testing, Moyer.
Mechanical Laboratory Methods of Testing Machines and Instruments, Smallwood.
Experimental Engineering and Manual for Testing, Carpenter and Diederichs.
Power Test Code, A. S. M. E.

REVIEW QUESTIONS AND PROBLEMS

1. What are the purposes for which tests are made on steam boilers?
2. Describe the preparations necessary to make a steam boiler test.
3. Name the conditions which should exist at start and finish of a steam boiler test.

4. How may the amount of air used during a boiler test be measured? Where this is not feasible, what method is used?

5. Prepare a data sheet suitable for the data pertaining to the weight of coal.

6. What data are shown on the boiler data chart?

7. Calculate all items which are printed in bold-face type for test No. 2, page 203.

8. Using the analysis for test No. 2, make a heat balance for the boiler, based on coal as fired.

9. During the test of a Babcock and Wilcox water-tube boiler at the Pennsylvania Navy Yard, the data were: water evaporated per hour, 13,552 lb.; oil used per hour, 871 lb.; quality of steam, 0.999; heat value of oil per lb., 19,525 B.t.u.; steam pressure, 300 lb. per sq. in. gage; barometer, 30.12 in. mercury; feedwater temperature at boiler, 199 deg. fahr. Find overall efficiency of the plant and factor of evaporation per pound of oil. How does the equivalent evaporation compare with that ordinarily obtained per pound of coal?

10. During a boiler test, the flue-gas analysis gave carbon dioxide, 15.10; oxygen, 3.09; carbon monoxide, 0.00; nitrogen by difference, 81.81. Find the weight of dry chimney gases per pound coal.

11. In Problem 10, the sum of the carbon dioxide and oxygen is 18.19 per cent. Give reasons why it is not 21 per cent. (See Art. 160, page 142.)

CHAPTER XI

PIPE SYSTEMS, PIPE, VALVES, AND PIPE ACCESSORIES

219. Foreword. — For the efficient and economical operation of a power plant, the layout of the piping system is of prime importance, and the care with which this is arranged will have its effect continuously on the operation of the plant. The points that should be considered in a piping system are:

1. Continuity of service and provision for extension.
2. Economical size and sufficient strength.
3. Expansion.
4. Drainage.
5. Support.

220. Division of Piping Systems. — The piping of a power plant may be conveniently considered under the following divisions:

1. *High-pressure piping.* — This includes the piping connecting the boilers with the engines, turbines and steam pumps. It includes the boiler leads, main steam header and auxiliary header, engine and turbine leads, connections to auxiliaries and low-pressure traps.

2. *Low-pressure piping.* — This includes all atmospheric exhaust lines, connections to feedwater heaters and exhaust steam heating.

3. *Vacuum piping.* — Includes all exhaust connections between the prime mover and condenser.

4. *Feedwater piping.* — Includes all connections to and from the water end of feed pumps and injectors and the feed lines of the boilers.

5. *Blow-off piping.*

6. *High- and low-pressure drainage systems.* — Includes drips, traps, and seals for the return of condensation either to the feedwater heater or direct to the boilers.

221. High-pressure Piping Systems. — The systems of piping employed for the main steam piping are:

1. Single-header.
2. Spider.
3. Loop, or ring.
4. Unit.
5. Duplicate.

Of these systems the one to be used depends upon the size of the plant and the character of the load. In plants having several boilers and en-

gines, the engine leads are connected to a common header which forms a flexible tie between the boilers and engines. This permits cutting any boiler into or out of service and re-distributing the load among the remaining boilers.

The **single-header system,** which is largely used in small and medium-sized plants, is illustrated in **Fig. 127,** with the boilers and engines ar

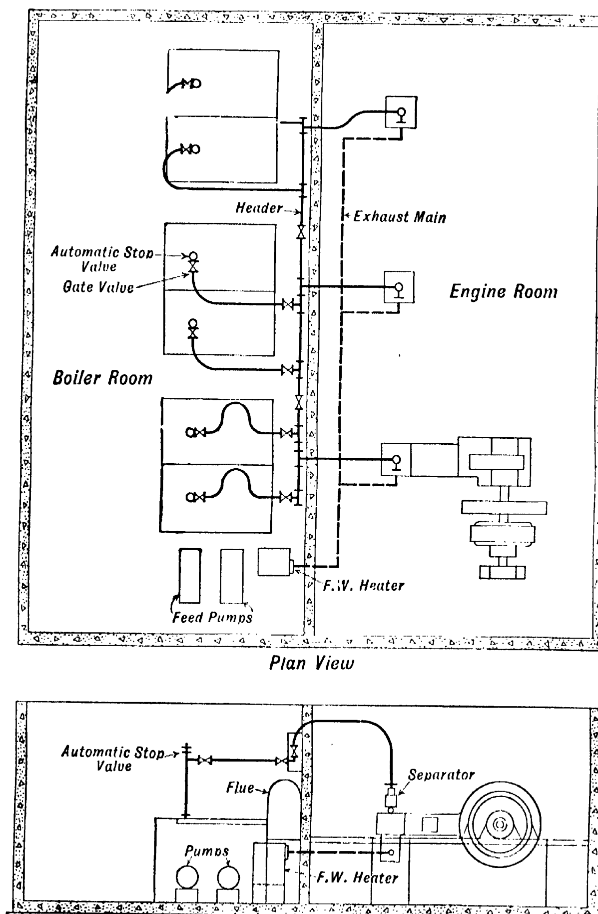

Fig. 127. — Single Header Pipe System.

ranged back to back. It has a low first cost and is simple and easy to extend. Two stop valves are located between the header and each battery of boilers, to permit cutting out any section when necessary. The header

is made slightly larger than the boiler lead or pipe connecting boiler and header. Several methods of providing for expansion are shown.

The **spider system,** Fig. 128, is simple and adaptable to small plants. The boiler leads are connected to a short header, long enough to contain

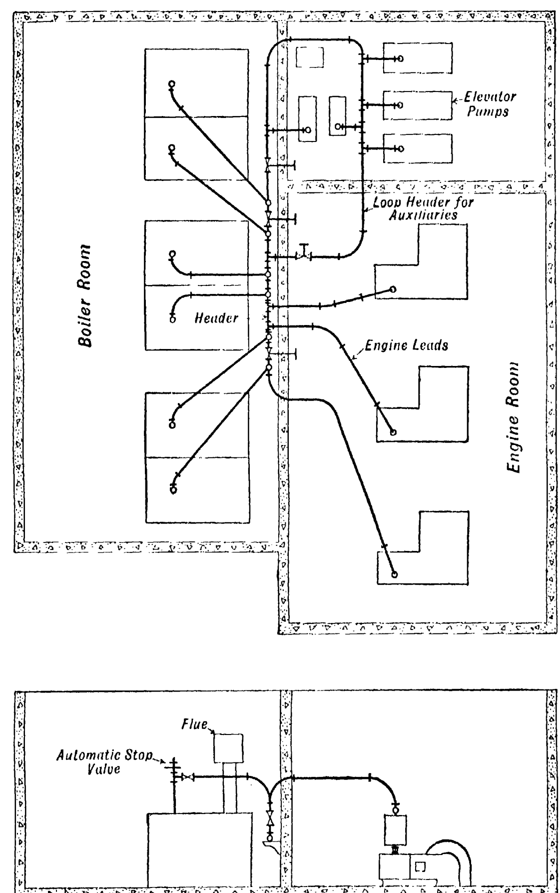

FIG. 128. Spider Pipe System.

the necessary valves and fittings. The short header reduces the danger of shutdown, reduces condensation, and is easily extended.

The **loop,** or **ring, system** has piping in which the header is arranged in the form of a loop, and hence requires a greater length of pipe than the single header and the spider systems. The valves in the header should be so

located as to permit cutting any section out of service and supplying steam from the remaining boilers. This system is not in general use, because of the high cost. It cannot be easily extended and has a large number of joints from which leakage may occur.

The **unit system** uses a single header, and each prime mover is supplied by its own battery of boilers, with the units cross connected to permit throwing any unit over on another battery of boilers. This system is used in large central stations and is frequently expanded to include a separate feedwater heater, pumps, economizers, condenser and chimney for each main unit.

The duplicate system has all piping installed in duplicate; it is not used to any extent, because of its high cost.

222. Boiler and Engine Leads. — An approved method of arranging the piping for boiler leads is shown in Fig. 129. By this arrangement

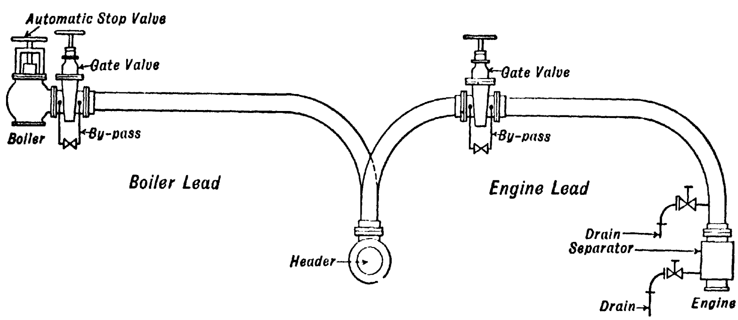

Fig. 129. — Boiler and Engine Leads.

pockets in which water might collect are avoided. Each boiler lead should have an approved automatic, quick-closing stop valve, which will close whenever the pressure in the header exceeds that in the boiler. In addition to this valve, a rising-stem gate valve should be used to permit closing the lead when making repairs. The rising stem shows at a glance whether the valve is closed or open.

Valves located on horizontal runs prevent water pockets, and this is the approved location of gate valves for all boiler and engine leads. The slope of the piping should be in the direction of steam flow, with the header connection at the upper side. Expansion in engine and boiler leads is provided for by long radius bends. The leads should be supported from above by hangers.

A typical engine lead is also shown in Fig. 129. It has a rising-stem gate valve and discharges into a receiver separator located just above the engine throttle valve. By this arrangement, known as the **triple swing connection,** the necessity of springing the bends into place is prevented,

because a swing adjustment can be made on three planes. It must have *a horizontal and two vertical joints* on one end of the connection.

223. Size of Leads and Headers. — Boiler and engine leads are ordinarily made one size smaller than the boiler nozzle or flange on an engine. The velocity of steam in the pipe is limited by good practice to 6,000 feet per minute. Velocities as high as 10,000 to 12,000 feet per minute are often used with turbines when the steam flow is constant, while 9,000 feet per minute is a maximum for steam-engine piping.

The size of a steam pipe, based on an allowable velocity, can be found by using the following equation:

$$A = \text{area of pipe in square inches} = \frac{144 \times W \times s}{V} \quad . \quad . \quad (72)$$

in which W = equivalent weight of steam flowing, pounds per minute.
s = specific volume of steam at absolute pressure in pipe, cubic feet.
V = velocity of steam, feet per minute.

Example 29. — Using the data given below, find the diameter of steam pipe required to keep the velocity of the steam to 6000 ft. per minute.

Solution. — Using Equation (72),

$$\text{Area in sq. in.} = \frac{144 \times W \times s}{V} = \frac{144 \times 260 \times 2.839}{6000} = 17.75 \text{ sq. in.}$$

W = 260 lb. per minute.
V = 6000 ft. per minute.
s = 2.839 cu. ft. per pound at 160 lb. per sq. in. abs., Table 7, page 90.

$$\text{Diameter of pipe} = d\sqrt{\frac{17.75 \times 4}{3.14}} = \sqrt{22.60} = 4.76 \text{ inches.}$$

Note: A 5-inch pipe would be used, as it would be the nearest commercial size.

In practice, the limiting factor, in determining the size of steam piping, is the allowable pressure drop, since the permissible velocities are governed by this drop. There are various formulae for computing the drop in pressure. **Babcock's equation** appears to give as satisfactory results as any:

$$P = 0.000131\left(1 + \frac{3.6}{D_1}\right)\frac{W^2L}{dD_1^5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

in which P = the difference in pressure between the two ends of the pipe in pounds per square inch.
W = weight of steam passing, pounds per minute.
d = mean density of steam, pounds per cubic feet.
D_1 = internal diameter of pipe, inches.
L = length of pipe in feet

Equation (73) may be written in the form

$$P = A\frac{W^2L}{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

in which A = a constant = $\dfrac{\left(1 + \dfrac{3.6}{D_1}\right) 0.000131}{D_1{}^5}$. Numerical values of A can be computed for the various sizes of steam pipes, and piping problems may thus be solved more conveniently.

Example 30. — A 5-inch steam pipe line has a length of 300 ft. and delivers 15,000 lb. of steam per hour, at a steam pressure of 165 lb. per sq. in. abs. Find the loss in pressure in the pipe line.

Solution. — Using Equation (74) with proper substitutions

$$P = \frac{A\ W^2\ L}{d} = 0.000,000,07 \times \frac{\overline{250}^2 \times 300}{0.363} = 3.62 \text{ lb.}$$

A for a 5-inch pipe = 0.000,000,07

$W = \dfrac{15,000}{60} = 250$ lb. per minute; $L = 300$ ft.

d = from steam table $= \dfrac{1}{s} = \dfrac{1}{2.757} = 0.363$ lb. per cu. ft.

The steam header for a single header system should be made equal in size to the largest boiler lead. If the engine and boiler are located at opposite ends of the header, the size may be computed by using a velocity of 8000 feet per minute for the rate of steam flow. The header should generally be carried on rigid supports and anchored to a support at the center of the header. *Rising-stem gate valves should be used between each battery of boilers.*

224. Exhaust Piping. — The exhaust piping connecting each engine or turbine with the exhaust steam main should have a gate valve to isolate the unit when making repairs. The use of pipe bends is not general. All piping should be short and direct, with a minimum number of joints which should be made tight to prevent leakage of air when attached to a condenser.

The size of exhaust piping is generally based on a velocity of 8000 to 9000 feet per minute for non-condensing operation; velocities as high as 20,000 feet per minute are permissible for condensing plants because the loss caused by friction is low.

225. Feedwater Piping. — The piping used for feedwater lines should be made of extra heavy steel or brass pipe, to guard against corrosion. Flanges made of cast steel give better service than cast-iron flanges. A feed, check, and stop valve are generally used for each feedwater connection to the boiler. The stop valve used should be of the globe pattern, as the gate valve cannot be used to regulate the flow, and pulsations from the feed pump cause it to clatter. The complete feedwater piping for a

non-condensing plant is shown in Fig. 130. In general, all equipment should be by-passed to permit the making of repairs without closing down the plant.

The flow of feedwater may be controlled by hand-operated valves or by a feedwater regulator which automatically maintains the water level

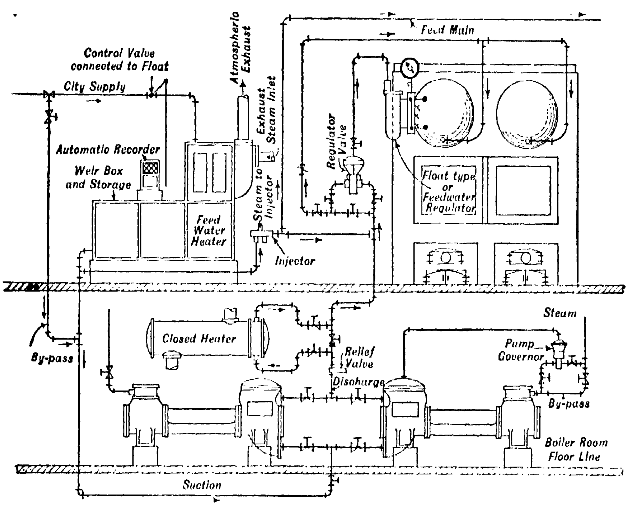

FIG. 130. — Non-condensing Feedwater Piping with Feed Pumps Installed in Duplicate.

in the boiler as nearly constant as possible. These regulators are not always positive in their action, and most engineers prefer to rely on hand regulation.

226. Feedwater Regulators. — Feedwater regulators are generally of the float or expansion type, and ordinarily operate in conjunction with a governor attached to the feedwater pumps. The regulator moves a valve located in the feedwater pipe and thus increases or diminishes the opening through which the feedwater flows. This action decreases or increases the pressure against which the pump is discharging, and the governor then changes the speed of the pump to feed the proper amount of water to maintain the water level.

The **Murray regulator,** Fig. 131, has a float-controlled valve mechanism. The chamber in which the float works is connected to the boiler by connections above and below the mean water level. The float is moved by variations in the water level and thus changes the pressure of the steam, which moves a balanced valve in the feedwater pipe and thus controls the supply of water. The balanced valve is normally held closed by a spring placed above a piston connected to the spindle of the valve. When a

heavy load comes upon the boiler, the steam pressure is slightly decreased and the water level raised, because of the increased evaporation. The float is thus raised and the steam pressure, acting on the under side of the regulating valve spindle, is lowered. The spring then partially closes the valve and decreases the flow until sufficient water has evaporated to lower the level, so that the regulator will again open and feed the boiler. A decrease in load lowers the water level because of decreased evaporation, and the float then operates to supply sufficient water to bring the water level normal.

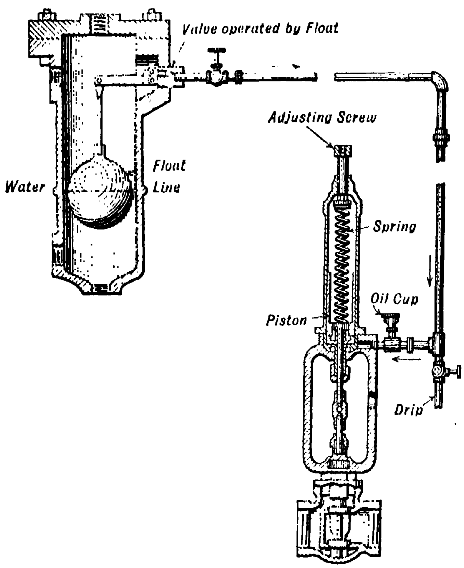

FIG. 131. — Sectional View of Murray Automatic Boiler Feed Regulator.

The **Copes regulator,** Fig. 132, is a simple type of regulator depending for its operation upon the expansion and contraction of an inclined tube. This tube is connected to the steam and water spaces of the boiler, in such a manner that when the water is at its lowest level it contains only steam and then has its maximum length. As the water level rises, water rises in the tube, which then contracts, the length of the tube depending upon the level to which

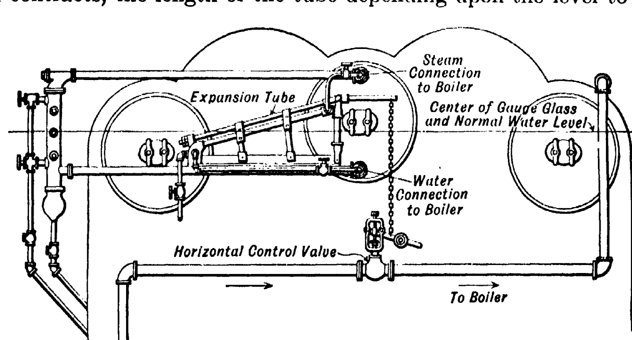

FIG. 132. — Improved Copes Expansion Type Feedwater Regulator.

it is filled with water. The tube is connected to a balanced valve in the feedwater line by a system of levers which move the valve as the height of the water level changes. With low water level, the tube is relatively long and the valve nearly open, thus giving maximum rate of flow; as the water rises, the tube is shortened and the rate of flow decreased.

227. High-pressure Drip Piping. — This piping automatically returns the steam condensed in the headers, steam separators, or other high-

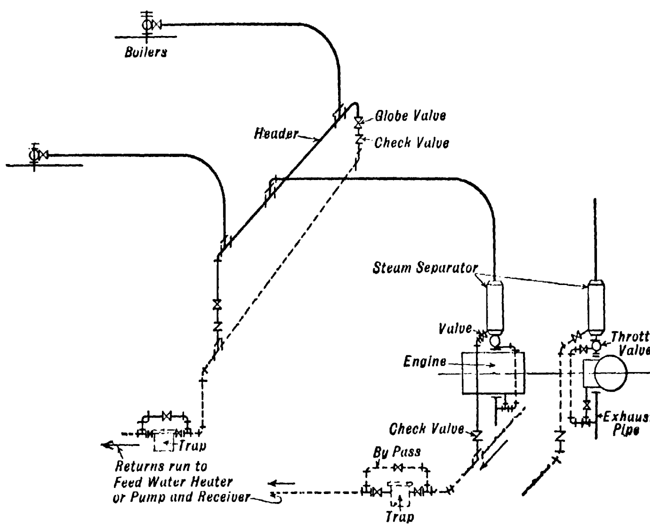

FIG. 133. — Arrangement of Drips for High Pressure Piping.

pressure piping to the boiler or feedwater heater, by means of **steam traps, pumps,** or the **steam loop.** An isometric sketch of a typical system is shown in Fig. 133. The size of header drips is usually from $\frac{3}{4}$- to 1-inch pipe, and that of throttle and engine drips $\frac{1}{2}$-inch pipe. *A check valve should be located in each drip when two or more drips are connected to the same trap.*

228. Steam Traps. — *Steam traps collect the water of condensation from steam apparatus, and automatically discharge it to a tank or hot well, with minimum loss of steam.* There are many types of steam traps, the most common of which are.

1. Bucket.
2. Float.
3. Tilting.
4. Differential.
5. Expansion.

The force of gravity causes the water of condensation to flow to the trap. Ordinarily, a steam trap will discharge against a pressure 5 pounds less than the steam pressure in the trap.

The **bucket trap,** Fig. 134, is the simplest type of trap, and is intermittent in its operation. It consists of an outside casting having a cover

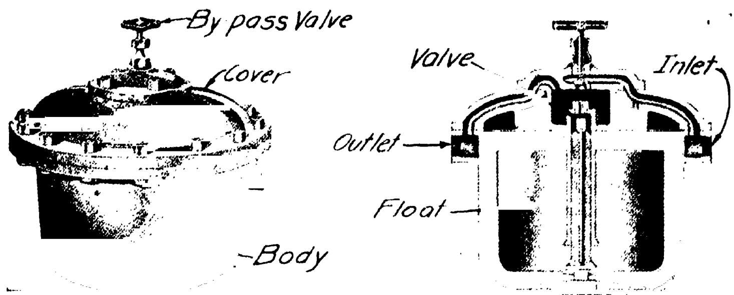

FIG. 134. — Bucket Steam Trap.

in which the steam and water passages are located. Inside the casting is a **bucket** which has a **needle valve** attached to it. The position of the bucket determines the position of the valve.

Water of condensation and steam enter the trap at the right and pass through the cored openings, in the cover, into the body of the trap. A **diaphragm** above the float diverts the water into the body of the trap. The water gradually rises, thus raising the float and closing the discharge valve. This valve remains closed until water rises in the body of the casting and overflows into the float in sufficient quantity to make the weight of the float, plus the water, great enough to sink the float and open the discharge valve. Steam pressure, acting upon the water within the float, then forces the water up through the sleeve which surrounds the valve and out at the left. This action continues until enough water is forced from the float to lighten the float sufficiently to allow it to rise and close the valve again. If for any reason it is desired to have the water of condensation pass without entering the trap, a small valve is provided in the cover. This valve controls a **by-pass** located on the cover.

A **float trap** is shown in Fig. 135, with all parts labeled. Water and steam enter at the inlet, and any sediment in the water is removed by the strainer. The water rises in the chamber until it seals the valve with 3 inches or more of water, the height of which is indicated by a gage glass. The float is then raised by the rising water, and the discharge valve is opened. The rate of discharge depends upon the position of the float, which is controlled by the amount of water entering. The valve is under water at all times. A small cam is provided to permit by-passing the

trap when necessary. The float is **counter-weighted** to permit using a smaller float. An **air valve** at the top is provided, for removing air from the trap, which will not work when air collects in the chamber. This type of trap gives a continuous discharge; but if it is not well made, leakage of steam is liable to occur. The high velocity of discharge also wears the valve seat.

The **tilting trap** is essentially a steel tank swung on a brass trunnion between two supports which are secured to a cast-iron bed plate. The supports consist of a valve connection on one side and water connection

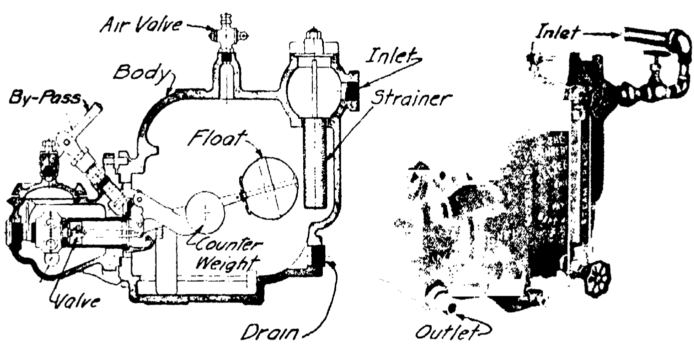

FIG. 135. — Anderson Float Trap.

on the other, both made of brass and shaped to form a suitable rest for the trunnion. A pet-cock located on top of the trap provides for exhausting the air from the tank. Buffers on each side limit the swing of the trap.

The tank is normally held horizontal by means of a weighted lever. When sufficient water has entered the tank to overcome the action of the lever and weight, the tank tilts backward. *This opens the discharge valve, and steam pressure forces the water from the tank until the weight on the lever again returns the tank to a horizontal position.* The weight is adjusted to allow sufficient water to remain in the tank to keep the outlet always water-sealed.

229. Return Traps. — The traps described above are of the **non-return type,** as the water is not returned to the boiler. There are traps that are located above the boiler level; when the trap is full, live steam at boiler pressure is admitted automatically to the trap. The pressure in the boiler and trap is thus equalized, and the water flows from the trap to the boiler by gravity. This type of trap is known as a **return trap.**

Steam traps are usually rated by the weight of water they will discharge per hour.

230. The Steam Loop. — A method of returning condensation from high-pressure lines to the boiler without using traps is shown in Fig. 136, and is known as the **Holly steam loop.** The figure illustrates the method used to return the high-pressure drips to the boiler located above a receiver to which the drips carry the condensation. The apparatus consists of a discharge chamber which is located at a considerable height

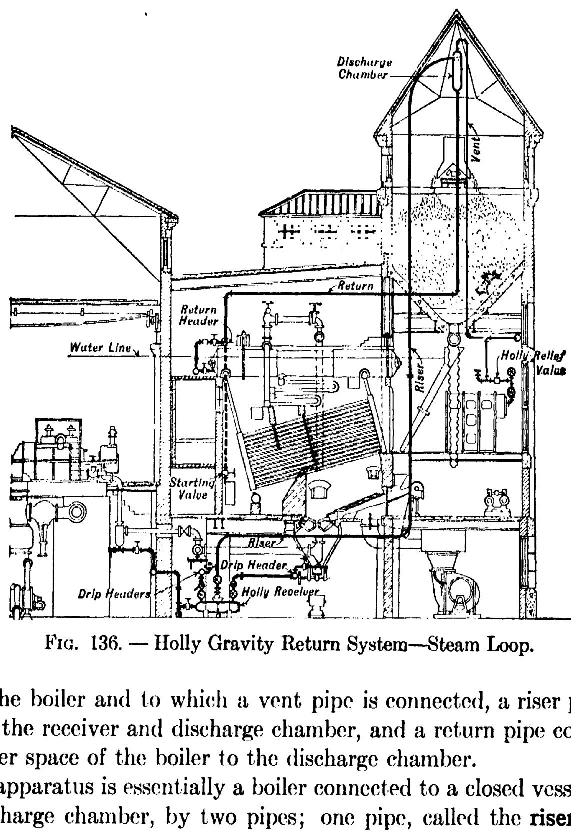

FIG. 136. — Holly Gravity Return System—Steam Loop.

above the boiler and to which a vent pipe is connected, a riser pipe connecting the receiver and discharge chamber, and a return pipe connecting the water space of the boiler to the discharge chamber.

This apparatus is essentially a boiler connected to a closed vessel, called the discharge chamber, by two pipes; one pipe, called the **riser,** is connected through the receiver to the steam space of the boiler, and the other, called the **return,** to the water space. Under this condition the water in the return stands at the same level as that in the boiler, because the pressure in the discharge chamber equals that in the boiler. If the pressure in the chamber is lowered by some suitable means, water will rise in the return to such a height that the added weight of water will balance the difference in pressure between the boiler and the chamber, and steam will flow through the riser in an effort to balance the pressure. Any

water carried by the steam will be swept along and delivered into the chamber where it falls to the bottom and enters the boiler through the return, as long as the added weight of water is greater than the difference in pressure between the boiler and the chamber.

The system shown in Fig. 136 operates as follows: the **starting valve** is opened until steam appears. It is then closed, and the **reducing valve** in the vent pipe is put into operation, thereby reducing the pressure in the discharge chamber and causing the condensation to be forced from the receiver to the discharge chamber, in the form of a spray produced by a series of holes in a pipe through which the condensation passes when leaving the receiver. The steam and entrained water are here separated, the steam passing through the vent pipe to the feedwater heater, and the water through the return pipe to the boiler. Once started, the process is continuous.

231. Commercial Piping. — Commercial pipe used in power plants is made of the following materials:

1. Wrought iron.
2. Mild steel.
3. Cast iron.
4. Brass.

Cast-iron pipe is used mainly for water and sewage systems; wrought-iron and steel pipe is used principally for steam, air and oil. *The size of pipe is stated in nominal inside diameter, up to and including 12-inch pipe. Above 12 inches, the size is based on the outside diameter, and the thickness of metal must be always given.* This pipe is known as **O. D. pipe.** The grades of steel pipe in general use are (1) **standard,** (2) **extra heavy,** and (3) **double extra heavy.** Table 22 gives the principal dimensions and areas of standard and extra-heavy pipe. *The outside diameter of standard, extra-heavy, and double extra-heavy pipe is the same for any given size; the inside diameters are different.* Extra-heavy pipe should be used for pressures above 125 pounds. A grade of pipe known as merchant pipe was formerly made in sizes up to 6 inches, but manufacturers do not carry it in stock at the present time. Most piping comes in random lengths and may or may not be threaded, the threading being done by means of **dies.** The type of die required for wrought iron does not give satisfactory threads when used on steel pipe, because it tears the threads.

The standard system of pipe threads is the **Briggs system.** The thread is given a taper of $\frac{3}{4}$ inch per foot, thus making it possible to secure a tight joint when screwing the pipe into a fitting.

232. Expansion. — *Whenever water, steam, or gas at high temperatures is to be conveyed by piping, proper provision must be made for expansion.* The movement between pipe lines may be taken up by **expansion joints, swing joints,** or **pipe bends.**

TABLE 22. — DIMENSIONS OF STANDARD AND EXTRA-HEAVY WROUGHT-IRON AND STEEL PIPE

Nominal Size	Diameter			Circumference			Internal Transverse Area	
	External Standard and Extra Heavy	Internal		External Standard and Extra Heavy	Internal		Standard	Extra Heavy
		Standard	Extra Heavy		Standard	Extra Heavy		
1/8	0.405	0.269	0.215	1.272	0.848	0.675	0.0573	0.0363
1/4	.540	.364	.302	1.696	1.144	.949	.1041	.0716
3/8	.675	.493	.423	2.121	1.552	1.329	.1917	.1405
1/2	.840	.622	.546	2.639	1.957	1.715	.3048	.2341
3/4	1.050	.824	.742	3.299	2.589	2.331	.5333	.4324
1	1.315	1.049	.957	4.131	3.292	3.007	.8626	.7193
1 1/4	1.660	1.380	1.278	5.215	4.335	4.015	1.496	1.287
1 1/2	1.900	1.610	1.500	5.969	5.061	4.712	2.038	1.767
2	2.375	2.067	1.939	7.461	6.494	6.092	3.356	2.953
2 1/2	2.875	2.469	2.323	9.032	7.753	7.298	4.784	4.238
3	3.500	3.068	2.900	10.996	9.636	9.111	7.388	6.605
3 1/2	4.000	3.548	3.364	12.566	11.146	10.568	9.887	8.888
4	4.500	4.026	3.826	14.137	12.648	12.020	12.730	11.497
4 1/2	5.000	4.506	4.290	15.708	14.162	13.477	15.961	14.454
5	5.563	5.047	4.813	17.477	15.849	15.121	19.990	18.194
6	6.625	6.065	5.761	20.813	19.054	18.099	28.888	26.067
7	7.625	7.023	6.625	23.955	22.063	20.813	38.738	34.472
8	8.625	7.981	7.625	27.096	25.076	23.955	50.040	45.664
9	9.625	8.941	8.625	30.238	28.089	27.096	62.776	58.426
10	10.750	10.020	9.750	33.772	31.477	30.631	78.839	74.662
11	11.750	11.000	10.750	36.914	34.558	33.772	95.033	90.763
12	12.750	12.000	11.750	40.055	37.700	36.914	113.098	108.43

NOTE. — Dimensions are normal and are in inches.

233. Expansion Joints. — The slip type of expansion joint, Fig. 137, may be a single or a double slip joint. It consists of a main casting which is anchored, a bronze sleeve which fits into this main casting and a **gland and packing** placed around the sleeve. The gland is adjusted to prevent leakage, by take-up bolts. The pipe line must be held in line, in order that this joint may work properly. Expansion and contraction move the sleeve in the main anchored casting. The number of expansion joints installed in a pipe line depends upon (1) the amount and direction of expansion and (2) the amount of expansion permitted by each joint.

In addition to the slip joint, a **corrugated copper expansion joint** is frequently used. It consists of a cylinder of corrugated copper held between two flanges. When used for high pressures the corrugations are reinforced with steel rings.

234. Pipe Bends. — Several typical expansion bends are shown in Fig. 138. The amount of expansion provided for by each bend depends upon the radius of the bend and the thickness of the pipe. The bend is placed at a suitable point in the pipe line and installed so that it will not act as a dam to obstruct steam or water flow. *Particular attention must*

be given to the proper drainage of the steam line when pipe bends are used. The bend, when cold, is customarily placed under initial tension equal to about one-half the total expansion; this decreases the amount of stress

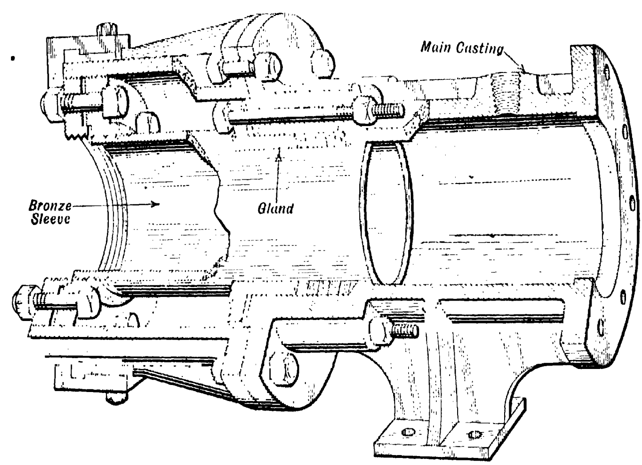

FIG. 137. — Ross Expansion Joint.

on the joint when hot. As an example of the amount of expansion of a typical bend: a 10-inch **U-pipe bend** will allow 3.2 inches expansion when bent to a radius of 80 inches.

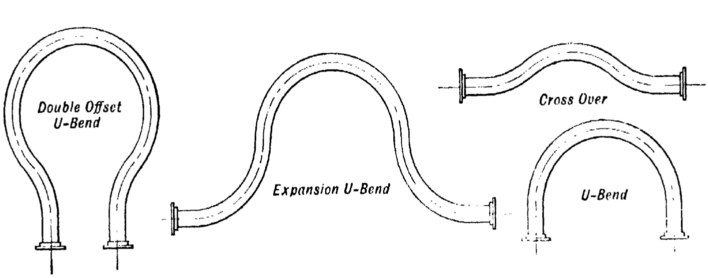

FIG. 138. — Typical Pipe Bends.

235. Amount of Expansion. — *The amount of expansion in piping depends upon the material from which the piping is made and the difference in temperature between the pipe when hot and when cold.* The amount of expansion in inches may be calculated by the following equation:

$$\text{Expansion in inches} = L \times C \times (t_1 - t_2) \times 12. \quad . \quad . \quad . \quad . \qquad (75)$$

in which L = length of pipe in feet.

C = coefficient of linear expansion per inch of length per deg. fahr.
= .00001111 for bronze.
= .0000068 for wrought iron.
= .0000067 for steel.
= .0000065 for cast iron.

t_1 = final temperature, deg. fahr.

t_2 = initial temperature, deg. fahr.

Example 31. — A steel steam pipe 293 ft. long carries steam at 190 lb. per sq. in. abs. pressure and superheated 125 deg. fahr. Find the theoretical amount of expansion on being heated from a room temperature of 80 deg. fahr.

Solution. — From Equation (75)

$$\begin{aligned}\text{Expansion in inches} &= L \times C \times (t_1 - t_2) \times 12\\ &= 293 \times 0.000{,}006{,}7 \times (502.6 - 80) \times 12\\ &= 9.95 \text{ inches.}\end{aligned}$$

L = 293 ft.; C = 0.000,006,7; t_2 = 80 deg. fahr.

t_1 = 377.6 + 125 = 502.6 deg. fahr., where 377.6 is found from the Steam Table at 190 lb. abs. pressure.

The actual expansion is somewhat less than the theoretical amount of expansion. It was formerly thought that the amount of expansion was about one-half of the theoretical, because the exterior surface does not reach the full steam temperature. It has, however, been demonstrated that, when the surface is well covered, the actual expansion is nearly the same as the theoretical expansion.

236. Pipe Fittings. — Commercial fittings used to join the various lengths of pipe are made in a variety of forms. *Screwed fittings are generally used in sizes up to $3\frac{1}{2}$ inches and flanged fittings for larger sizes.*

The material from which fittings are generally made is cast iron; but malleable iron, steel, steel alloy, and brass fittings are also used. *The material depends upon the service for which it is to be used.* Fittings may be classed as: (1) **low pressure fittings,** for pressures around 25 pounds per square inch; (2) **standard fittings** for 125 pounds per square inch; and (3) **extra heavy fittings** for pressures of 250 pounds per square inch.

237. Screwed Fittings. — The various screwed fittings are illustrated in Fig. 139. **Nipples,** or short pieces of threaded pipe, **couplings, elbows, return bends, T's, crosses, laterals, Y-branches, bushings, and reducing fittings** are used to join pipe lengths. **Plugs** and **caps** are used to close the ends of fittings and pipes. Reducing bushings and fittings may be tapped eccentrically to permit free drainage.

Unions, Fig. 140, are used to provide a means of disconnecting a pipe line. They are seldom used above 4 inches nominal diameter. The parts of the union are the **nut, male end,** and **female end.** The seat is usually made of brass to prevent rusting

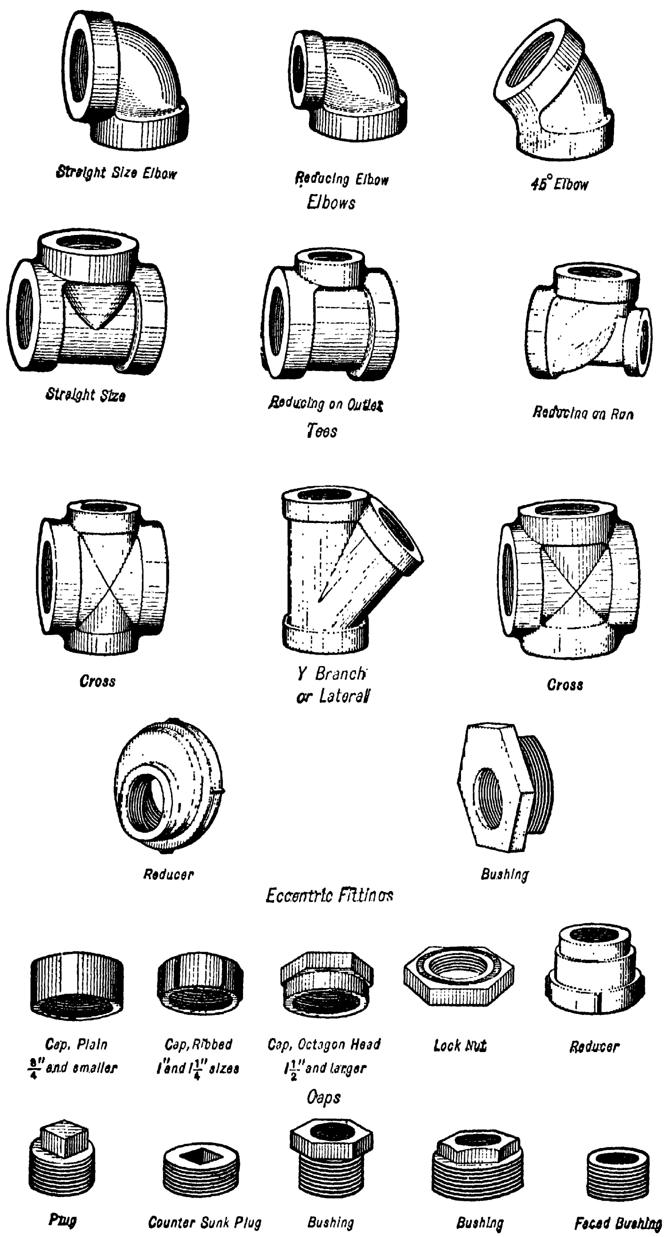

FIG. 139. — Cast-iron Fittings.

$$\text{Expansion in inches} = L \times C \times (t_1 - t_2) \times 12 \quad . \quad . \quad . \quad . \quad . \quad (75)$$

in which L = length of pipe in feet.

C = coefficient of linear expansion per inch of length per deg. fahr.
= .00001111 for bronze.
= .0000068 for wrought iron.
= .0000067 for steel.
= .0000065 for cast iron.

t_1 = final temperature, deg. fahr.

t_2 = initial temperature, deg. fahr.

Example 31. — A steel steam pipe 293 ft. long carries steam at 190 lb. per sq. in. abs. pressure and superheated 125 deg. fahr. Find the theoretical amount of expansion on being heated from a room temperature of 80 deg. fahr.

Solution. — From Equation (75)

$$\begin{aligned}\text{Expansion in inches} &= L \times C \times (t_1 - t_2) \times 12 \\ &= 293 \times 0.000,006,7 \times (502.6 - 80) \times 12 \\ &= 9.95 \text{ inches.}\end{aligned}$$

L = 293 ft.; C = 0.000,006,7; t_2 = 80 deg. fahr.

t_1 = 377.6 + 125 = 502.6 deg. fahr., where 377.6 is found from the Steam Table at 190 lb. abs. pressure.

The actual expansion is somewhat less than the theoretical amount of expansion. It was formerly thought that the amount of expansion was about one-half of the theoretical, because the exterior surface does not reach the full steam temperature. It has, however, been demonstrated that, when the surface is well covered, the actual expansion is nearly the same as the theoretical expansion.

236. Pipe Fittings. — Commercial fittings used to join the various lengths of pipe are made in a variety of forms. *Screwed fittings are generally used in sizes up to 3½ inches and flanged fittings for larger sizes.*

The material from which fittings are generally made is cast iron; but malleable iron, steel, steel alloy, and brass fittings are also used. *The material depends upon the service for which it is to be used.* Fittings may be classed as: (1) **low pressure fittings,** for pressures around 25 pounds per square inch; (2) **standard fittings** for 125 pounds per square inch; and (3) **extra heavy fittings** for pressures of 250 pounds per square inch.

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the flange is bored smaller than the pipe. It is then heated and placed on the pipe, after which the pipe is expanded to fit corrugations in the hub of the flange, and the surface is then faced smooth. This joint is satisfactory for high-pressure steam. The **Vanstone or lap joint** is the best type of commercial joint for high pressure and temperature; it is dependable and requires no attention besides occasionally replacing a gasket. The pipe is rolled over against the flange at right angles to the axis of the pipe. The lap is then faced on front and edge, and acts as a bearing for the gasket when making the joint, while the flanges act as swivel collars to hold the pipe together. This flange is adaptable to all classes of service, for pressure up to 1000 pounds per square inch or above. **Welded** flanges are made of forged steel welded to the pipe and afterward faced to a true face. This joint costs much more than the other types of joints.

239. Methods of Facing Flanges. — In order to make a tight joint the faces of the flanges are machined in the following ways: (1) plain straight face, (2) raised face, smooth finish for gasket, (3) raised face, finished for ground joint, (4) tongue and groove, (5) male and female, (6) plain face corrugated or scored, (7) ball shape for ground joint.

The plain straight face has the entire face of the flange machined. A gasket, which should be fairly thick to make a tight joint, is placed between the flanges. This joint is good for pressures below 125 pounds per square inch.

The raised face, smooth finish for gasket, is made by raising the face of the flange between the bore and inside of the bolt holes from $\frac{1}{32}$ to $\frac{1}{16}$ inch above the remainder of the flange. It makes a satisfactory joint, is suitable for high steam pressures, and is the most common type of joint. A ring gasket is used between the flanges.

The raised face, ground joint, is similar to the one employed when gaskets are used, except that the face is machined perfectly true and a gasket is not required.

The tongue and groove joint is made by machining a circular ring on one flange and a corresponding recess on the other. It is a popular type of joint; but it is more expensive and is more difficult to disassemble, when replacing a valve or fitting than the previous types. For ammonia and hydraulic pipe lines there is no better joint.

The male and female joint is expensive to install and disassemble. It is adapted to high pressures, as it prevents the gasket from blowing out.

The plain face, corrugated, and scored joints are similar. In the former the face has concentric circles cut in the face by a round-edged tool. The scored joint has concentric rings made with a sharp-edged tool. This joint is good for oil and acid lines.

The ball-shaped flange joint resembles the screwed union joint. Gaskets are eliminated and misalignment of the pipe line is taken care of by the spherical joint.

$$\text{Expansion in inches} = L \times C \times (t_1 - t_2) \times 12. \quad . \quad . \quad . \quad . \qquad (75)$$

in which L = length of pipe in feet.

C = coefficient of linear expansion per inch of length per deg. fahr.
= .00001111 for bronze.
= .0000068 for wrought iron.
= .0000067 for steel.
= .0000065 for cast iron.

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Solution. — From Equation (75)

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Unions, Fig. 140, are used to provide a means of disconnecting a pipe line. They are seldom used above 4 inches nominal diameter. The parts of the union are the **nut, male end,** and **female end.** The seat is usually made of brass to prevent rusting

rising spindle type the spindle screws into the wedge as the wheel is revolved, in which case it is not apparent to the eye whether the valve is closed or open. The wedge may be solid or split, and should have a slight

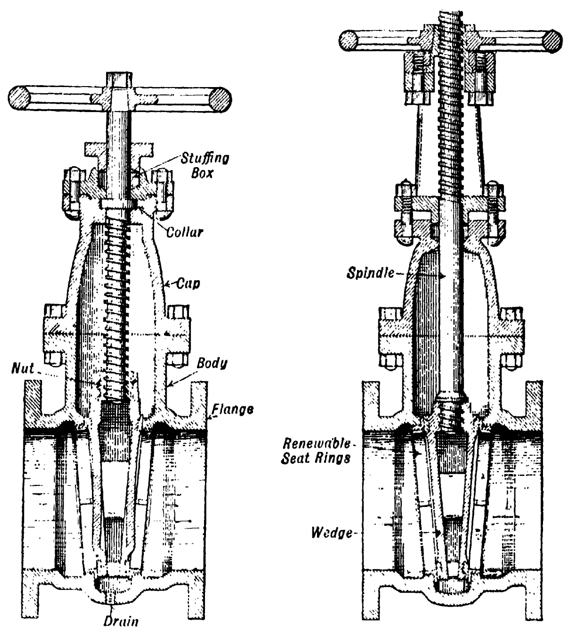

FIG. 143. — Non-rising and Rising Spindle Gate Valves with Iron Body and Brass Mountings.

taper, so that it will close tight when pushed down by the spindle. The top of the wedge is made to seat against the bonnet when wide open, which permits packing the stuffing box with the valve open.

Small sizes up to 3 inches are made of brass and usually have screwed ends, Fig. 144. The bonnet, which is screwed to the body by a **union nut,** carries a stuffing box through which the spindle rises and to which the gland is held by a nut. A wheel is attached to the top of the spindle, and a shoulder and hub on the lower end of the spindle turn in a split wedge which seats against the tapered faces. The top of the shoulder is finished and, when open, bears against the bottom of the bonnet and permits packing the valve under pressure.

244. Globe and Angle Valves. — Large sizes of these valves are made of cast iron with brass mountings. The construction is similar to that of the gate valve. except that the seat is horizontal, and the valve disk is

$$\text{Expansion in inches} = L \times C \times (t_1 - t_2) \times 12 \quad . \; . \; . \; . \; . \quad (75)$$

in which L = length of pipe in feet.

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closes, because the pressure below the disk is lower than on the header side. The disk remains closed as long as the pressure in the boiler is below that in the header and can only be opened with pressure on the boiler side. The valve should not stick, chatter, or hammer while performing its function. A dash pot is fitted to the upper end of the disk spindle to prevent chattering or violent movement. The outside stem is not attached to the valve disk, but the valve can be closed like any stop valve by screwing the handwheel down. The disk opens on a difference of pressure of one pound. Valves of this type are often arranged to close by steam pressure when desired.

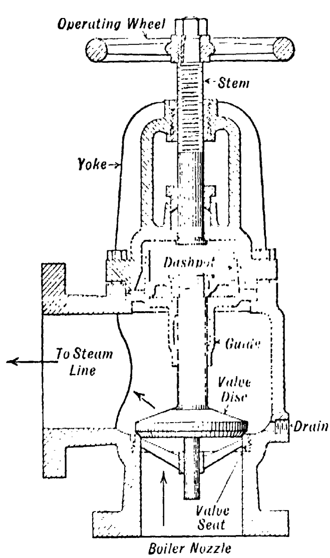

FIG. 147. — Foster Automatic Non-return Stop Valve.

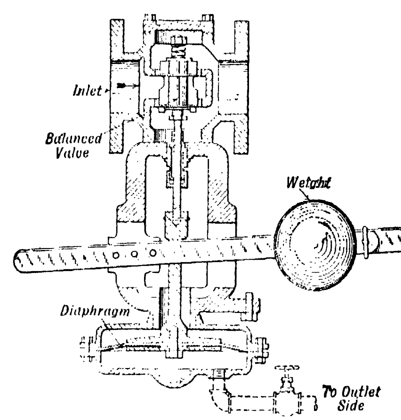

FIG. 148. — Kieley Lever and Weight Reducing Valve.

246. Reducing Valve. — This valve is used where it is desired to reduce the steam pressure, as for use in a heating system. Figure 148 illustrates the construction of a weight-and-lever type of reducing valve, consisting of a body connected into the pipe line. In the body is a balanced disk valve attached to a spindle, to the lower end of which a diaphragm is attached; just above the diaphragm is a lever, which is pivoted on an extension of the valve body. High-pressure steam enters from the left and passes through the balanced valve, which is normally held open by a weight located on the lever. The low-pressure side of the valve is connected by a pipe to the chamber below the diaphragm, in order to have the low-pressure steam act on the under side of the diaphragm. When the steam pressure acting on the diaphragm builds up sufficiently to lift

the weight, the spindle is raised, the balanced valve partially closed, and the pressure reduced to a value that will just balance the weight. The position of the weight determines the lower pressure.

247. Pipe Coverings. — *Coverings made from heat insulating material, such as magnesia or asbestos fiber, should be placed on all air, water and steam piping, valves and fittings.* Cold-water pipes are often covered, to prevent sweating in hot weather. The efficiency of pipe coverings may be as high as 86 per cent.

248. Pipe Hangers, Rollers and Supports. — Pipe hangers, Fig. 149, are used to relieve fittings of strain and keep a pipe line from sagging.

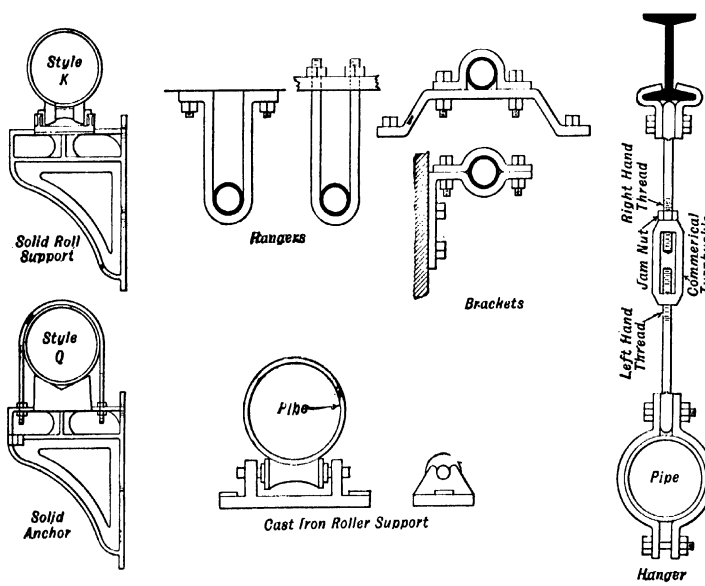

FIG. 149. — Typical Pipe Hangers, Brackets and Roller with Frame.

They are ordinarily attached to the framework of the building, and are free to swing as the pipe expands. In addition to hangers, pipes are supported on rollers which permit the pipe to expand and contract, and which are carried by a roller frame attached to a bracket located on the wall or frame of the building. When it is necessary to support pipes at a considerable distance from the floor, a pipe column is used.

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Steam Power Plant Engineering, GEBHARDT.
Mechanical Equipment of Buildings, Vols. I and II, HARDING and WILLARD.
Mechanical Engineers Handbook MARKS.

Handbook on Piping, SVENSON.
Power Plant Engineering, FERNALD and ORROK.
Steam, BABCOCK and WILCOX Co.

REVIEW QUESTIONS AND PROBLEMS

1. Name five requirements which a well-designed pipe line should satisfy.
2. Sketch the layout of (a) a single-header system of piping, (b) a spider system.
3. Sketch a proper arrangement of piping for a boiler lead.
4. A 1700 i.hp. engine, using 12.5 lb. of steam per i.hp. per hour, has a supply pipe 40 ft. long from the header. Average steam pressure, 160.3 lb. per sq. in. gage; average amount of superheat, 40 deg. fahr. Pressure drop from header to engine, 1.0 pound. What should be the diameter of the supply pipe?
5. Describe a typical feedwater piping system, using sketch.
6. What is a steam trap? Name four types and describe one type mentioned.
7. What is a return trap? Describe the operation of a steam loop.
8. Name the grades of steam piping.
9. Name three methods of providing for the expansion of a steam line.
10. A 5-inch steel pipe line, 600 ft. long, carries steam at an average pressure of 155.3 lb. gage, and average superheat in the line, 50 deg. fahr. Temperature before turning on steam, 50 deg. fahr. Find the allowance that must be made for expansion if the pipe is straight. Barometer, 29.92 in. mercury.
11. Name ten pipe fittings, and state the function of each.
12. Name two types of stop valves. Describe a non-return stop valve.
13. Describe a reducing valve and state why such a valve is used.
14. What is the purpose of pipe covering?

CHAPTER XII

FEEDWATER HEATERS AND PURIFIERS. FEEDWATER PURIFICATION. BOILER SCALE. CORROSION. TUBE CLEANERS AND SOOT BLOWERS

249. Foreword. — Feedwater heaters are used primarily for saving heat and thus increasing the economy of the boiler plant, also for purifying the water, since some impurities are separated from the water when it is heated.

Incidentally, increasing the temperature of the water tends to reduce strain caused by difference in temperature in the parts of the boiler and thus prolongs the life of the boiler. With large heaters the storage of water is an advantage, since a considerable quantity of hot water may be ready to supply a sudden demand.

For the best conditions of operation, the temperature of the feedwater should be nearly equal to the temperature of the water in the boiler, and its quality should be such that it will not cause corrosion of the boiler or a deposit of scale.

At ordinary boiler pressure, an increase of 10 to 12 deg. fahr. in the temperature of the feedwater corresponds to about 1 per cent gain in the efficiency of the boiler plant. A deposit of scale not only prevents the ready transfer of heat but endangers the life of the boiler by allowing the plate or tubes to become overheated.

250. Classification of Feedwater Heaters. — Feedwater heaters may be classified according to:

1. Source of heat.
2. Method of transmission of heat (contact or conduction).
3. Pressure of steam.
4. Arrangement for flow of the steam.

(1) Feedwater may be heated by exhaust steam, by hot waste gases, or sometimes by live steam; that is, steam taken directly from the boiler. If exhaust steam supplies the heat the apparatus is called a **feedwater heater,** while if hot gases furnish the heat it is called an **economizer.** A heater using live steam is often called a **purifier,** although all heaters are purifiers to some extent.

(2) If steam comes directly into contact with the water, the heater is an **open-type heater.** If heat is transmitted through walls which separate the steam from the water, the heater is of the **closed type.**

(3) Closed heaters, through which the exhaust steam from an engine passes on its way to the condenser, are called **primary heaters,** because cold feedwater sometimes receives heat first in this way. Steam exhausted at or near atmospheric pressure is used to heat water in open or closed heaters, which are called **secondary,** or **atmospheric, heaters.**

(4) The heater is of the **induced type** when only that portion of the steam which is induced enters the heater, but all the steam entering is condensed, thus lowering the pressure slightly and drawing more steam in. It is called a **through heater** when all of the steam is forced through the heater and only a part of the steam is condensed, the remainder being discharged.

251. Open Feedwater Heater. — A section of a Cochrane open feedwater heater and receiver, with part of the shell cut away to give a view of the interior, is shown in Fig. 150. It is made of cast-iron plates bolted together, is provided with doors to allow cleaning, and is intended for pressures as high as 10 pounds per square inch.

This heater is called a "steam stack cut-out" heater and is arranged for use when more exhaust steam is available than is required for heating the feedwater.

Part of the steam enters the heater through an opening controlled by a **cut-out valve,** the remainder passing directly to the exhaust outlet. A large, fluted oil-separator, in the exhaust inlet to the heater, removes the oil from the exhaust steam.

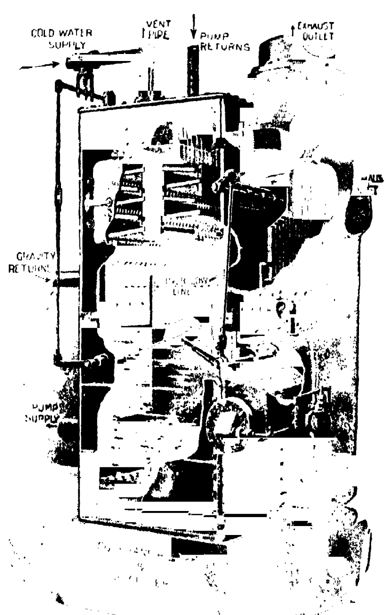

FIG. 150. — Cochrane Open Type Steam Stack Cut-out Feedwater Heater.

Cold water enters a reservoir at the top and overflows to a series of **corrugated copper trays,** or pans, and thence to the filter compartment below. While falling from the reservoir and pans, the water is so subdivided into sprays that it offers a large amount of surface to the entering steam, which is immediately condensed by direct contact. The quantity of cold water entering the heater is controlled by a **float** in the tank, which operates an inlet valve in the supply pipe.

The hot water passes down through the **coke filter** and enters the compartment at the left, where a section of the partition is shown cut away to reveal the outlet to the feed pump. The heater is flooded to the overflow level, and thus floating impurities pass off into the **float trap** at the right and are discharged to the waste pipe, by action of the trap. The discharge from the oil-separator also enters this trap and is discharged.

When the " steam stack cut-out " valve is closed, the overflow outlet is also closed, and no steam can enter the heater. This permits cleaning the heater while the engine is in operation.

A considerable amount of the soft scale-making material in the water is deposited on the trays and may be removed by opening the cast-iron doors of the heater and taking out the trays.

This heater serves as a receiving tank or hot well, to which hot condensate is returned by gravity or by means of a pump. A **vent pipe** is pro-

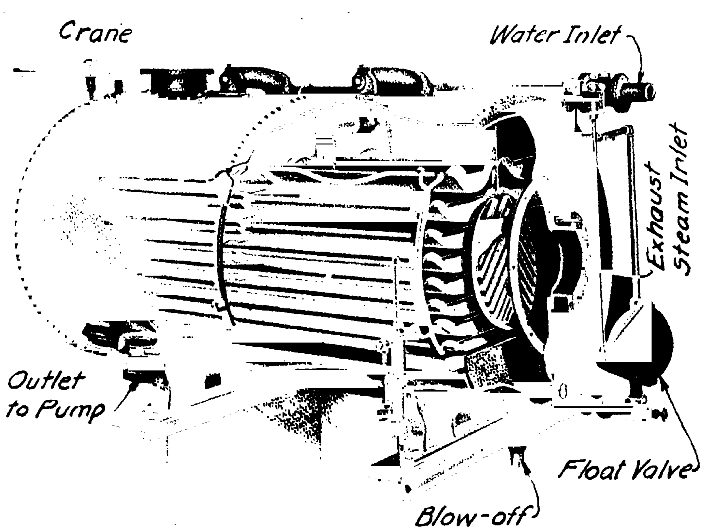

FIG. 151. — Hoppes Horizontal Open Type Feedwater Heater

vided to discharge the gases that are freed, and to ensure low pressure in the heater. The heater is located on the suction side of the pump.

The **Hoppes horizontal open feedwater heater,** Fig. 151, illustrates the pan type of heater. The water entering through the supply pipe is controlled by a float valve. It overflows from pan to pan, receiving heat from the exhaust steam, and is drawn away by the pump, which is generally placed below it. The exhaust steam passes over an oil-separator before entering the space between the trays. After considerable scale has ac-

cumulated the heater head can be removed and the trays taken out and cleaned. A crane is provided to assist in swinging the head out of the way and back into position.

252. Closed Heater. — This type of heater is connected into the feed line between the pump and the boiler. Such heaters are classified, according to the position and direction of movement of the steam and water, as (1) water-tube, or (2) steam-tube, either type having parallel currents or counter currents; that is, with steam and water moving in the same direction or in opposite directions. In **single-flow** water-tube heaters the water flows in one direction only; in a **multi-flow** heater the water passes back and forth in its passage through the heater tubes. A **coil heater** has one or more coils of pipe carrying the water.

A vertical closed heater of the multi-flow water-tube type, cut away at the top and bottom to show its construction, is illustrated in Fig. 152. The shell is made of cast iron and the tubes of corrugated copper, expanded in the tube heads and supported by a thimble. The corrugations allow for expansion and contraction with change of temperature, increase the heating surface, and, by agitating the water in its passage, tend to bring cooler particles of the water stream against the heating surface, and thus increase the efficiency of heat transfer. The irregular movement of the water and the movement of the tube by expansion and contraction also tend to prevent deposit of scale on the tubes. Exhaust steam enters against a baffle plate, used to protect the tubes from impact of the steam, and is distributed to the top and bottom. Condensed steam is discharged through the **drain pipe** shown at the left of the figure. Feedwater enters near the

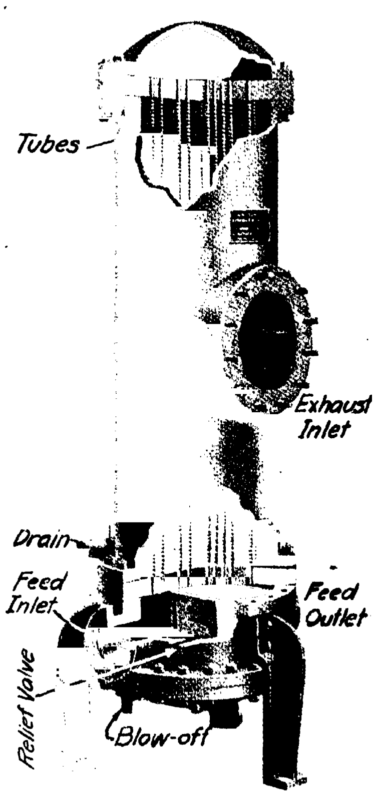

FIG. 152. — Wainwright Corrugated-tube Closed Feedwater Heater.

bottom, passes to the left top compartment, thence to the center bottom and right top compartments, and down to the feed outlet. There are two **partition plates** at the bottom and one at the top to direct the flow. A blow-off connection is shown and, as this heater is under boiler pressure or slightly higher, when the pump is working, provision is made for connecting a relief or safety valve.

A spiral coil is sometimes used, in straight brass tubes, for agitation of the water. This type of tube is expanded into one tube sheet and provided with a deep packing box and screwed gland in the other tube sheet, to allow for expansion and contraction.

253. Coil Tube Heater. — A multi-coil heater is illustrated in Fig. 153, in which a Reilly through type of heater, with part of the cast-iron shell

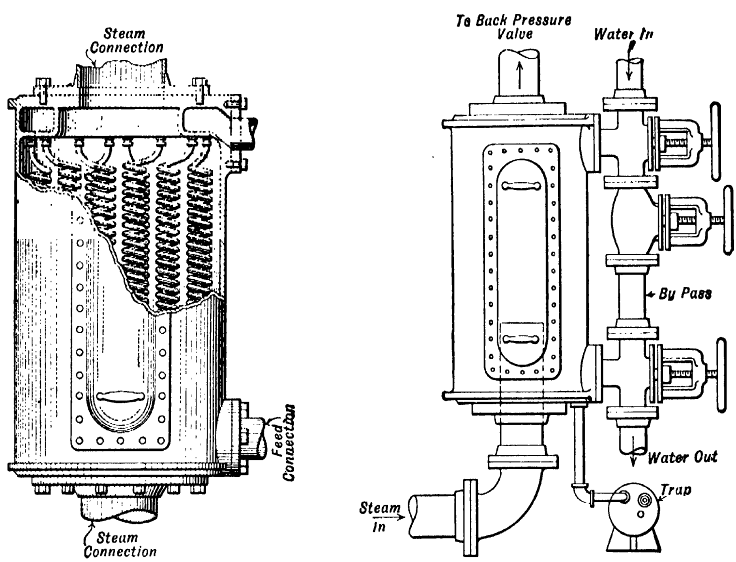

FIG. 153. — Reilly Multi-coil Exhaust Steam Heater and Piping Connections for through Type of Heater.

removed, shows the coils and their connections. The coils are interchangeable and are connected to the headers at top and bottom by metal screwed union joints. Expansion is taken care of by the form of the coils.

The arrangement of piping for a through coil heater is also shown in Fig. 153. The pressure of the steam passing through the heater is kept constant by the weighted back-pressure valve. By closing the upper and lower valves and opening the middle, or by-pass valve, the heater can be taken out of service to allow inspection of the coils. A trap is connected

as shown, to receive the steam condensed in the heater. All heaters should be by-passed to allow for cleaning.

254. Advantages and Disadvantages Claimed for Open and Closed Types of Feedwater Heaters. — In general, the advantages of one type are the disadvantages of the other type.

Open Type of Feedwater Heater. — The advantages are:

1. *Saving in heat.* — Direct contact with steam and water gives efficiency in heat transmission, and less exhaust steam is required to heat the water through the same temperature range.
2. *Saving in water.* — The condensed steam is ready to be used, instead of requiring an equal quantity of cold water. This amounts to about one-seventh of all the water that would have to be supplied to a closed heater for the same total amount of feedwater.
3. *Improvement in quality of the whole feedwater supply.* — The condensed steam is practically pure water. The temperature in the heater is sufficient to liberate gases and air in the feedwater which cause corrosion and pitting in the boiler, and to precipitate some scale-making material out of the water. Impurities such as mud are removed by the filter, and lighter flocculent impurity is floated off over the waste discharge.
4. *Serves as a hot well.* — It may be used as a hot well to receive returns from the heating system.
5. *Easy to clean.* — Easy and rapid cleaning is possible, as all parts of the heater are accessible.

The principal disadvantage of the open heater is the opportunity allowed for oil to get into the feedwater, due to inefficient oil-separators and lack of care in cleaning the filter, though it should be stated that efficient oil-separators are furnished by the makers and are in common use.

Closed Exhaust Steam Heater. — The advantages of this type of heater are:

1. *No oil.* — There is no danger of oil entering the heater from the engine exhaust. This is often a controlling factor in the choice of a heater.
2. *Improvement in quality of feedwater.* — Some scale-making material is precipitated and may be discharged.

The disadvantages are: A safety valve is required to prevent undue rise in pressure, and a tube failure may cause trouble because the tubes are under pressure. In general the advantages of the open heater are the disadvantages of the closed, except as noted.

255. Live Steam Heaters and Purifiers. — The principal object in the use of this type of heater is the purification of the water from hard scale-making material. At a temperature of about 300 deg. fahr., the sulphates of lime and magnesia, which make a hard scale, are precipitated. These sulphates are not affected by low-temperature steam. Some gain

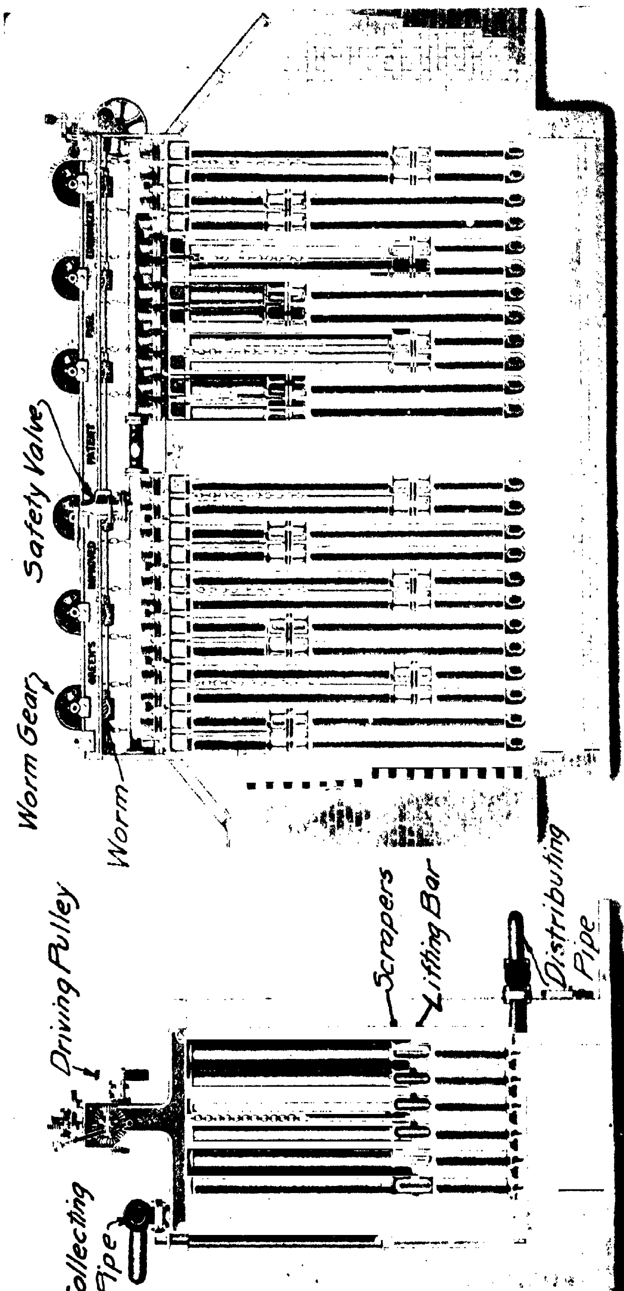

Fig. 154. — Side and End View of Green Fuel Economizer

in efficiency in heating feedwater by the use of live steam heaters has been shown, an explanation for which was offered in *Steam* * by the suggestion that heat may be transferred more readily at considerable temperature differences.

The live steam purifier is of the pan type and in construction resembles the Hoppes heater (Fig. 151, page 232). It is, however, operated at boiler pressure instead of at exhaust pressure. Water is delivered to the top pan and runs over the sides to the pans below; it then enters the boiler by gravity, because the location of the heater is above the water level. Air and gas expelled from the water must be vented to some high-pressure steam-pipe line, to prevent the heater from becoming air bound.

256. Fuel Saved by Heating Feedwater. — For the sake of economy, exhaust steam should be used for heating feedwater, wherever possible. The heat in the exhaust of an engine or turbine is a large proportion of the heat supplied. All the heat, except that used in work and lost by friction, radiation and leakage, can be returned to the boiler in the feedwater. If an engine using steam at 150 pounds per square inch gage pressure exhausts into a heater at atmospheric pressure, and the feedwater enters the heater at 50 deg. fahr. and leaves at 210 deg. fahr. the exhaust steam furnishes 159.01 B.t.u., which is about 14 per cent or nearly one-seventh of the total heat, 1178.9 B.t.u., required to evaporate one pound of water under the conditions. Thus the capacity of the boiler plant is increased without increasing the number of boilers; or it will be possible to furnish the same amount of steam with fewer boilers and less coal.

257. Fuel Economizers. — The Green fuel economizer, shown in Fig. 154, is an example of a feedwater heater receiving heat from waste gases. It consists of a series of vertical cast-iron pipes 9 to 12 feet long and $4\frac{9}{16}$ inches external diameter, arranged in **parallel rows** across the flue and connected into cast-iron headers at top and bottom. The bottom headers are connected by flanged joints at the front side to bottom branch distributing pipes. The top headers are similarly connected at the rear to the top branch collecting pipes. The distributing and collecting pipes, for the groups of vertical pipes making up the economizer, are connected by U-shaped, cast-iron, flanged fittings to allow for expansion and contraction. The tubes have metal-to-metal taper fits into the headers at both ends and are forced into place by hydraulic pressure. Opposite the top of each tube end in the top header is a hand-hole, closed by a lid having a conical seat and held in place by a yoke and bolt. The lids are removed through one opening which is made larger than the rest.

Feedwater enters the bottom branch pipe at the end nearest the point where the gases leave the economizer and, after passing through headers and pipes, leaves the top branch pipe at the end where the gases enter,

* Babcock and Wilcox Co.

thus ensuring the hottest water for boiler-feed. A safety valve is located on a flanged nozzle on the top branch pipe, next to the exit connection to the boiler-feed line. A bottom blow-off valve is connected to the bottom branch pipe to discharge sediment.

Scrapers for removing the soot, which falls to a space below the pipes, are continuously moved up and down the pipes by chains connected to a power-driven mechanism located above the economizer. The scrapers for two pipe sections are carried by a **lifting bar** and are held in place on the down stroke by a guard bolted to the bar. A chain from the lifting bar passes up between the headers, over a grooved pulley and down to a similar lifting bar and set of scrapers on the two adjoining tube sections. Thus, the weight of one set of pipe scrapers balances the weight of the other set.

The **reversing mechanism** has a large bevel gear fastened to the shaft running lengthwise above the economizer. This operates the chain pulleys by means of worms and worm gears. The direction of rotation of the shaft is reversed by a lever which is thrown first to one side and then to the other by the action of fingers attached to an idler wheel, which is operated by a worm gear and wheel from the shaft. The lever slides a toothed clutch keyed on the cross shaft, making the small bevel gears, located at each end of the cross shaft, alternately drivers and idlers.

The economizer is enclosed in a brick setting or, in more modern construction, in an asbestos-lined sheet-metal casing. A **by-pass** with dampers is provided for the gases, to allow for cleaning and repair, and a by-pass is also arranged in the feedwater piping, as for any heater. An illustration

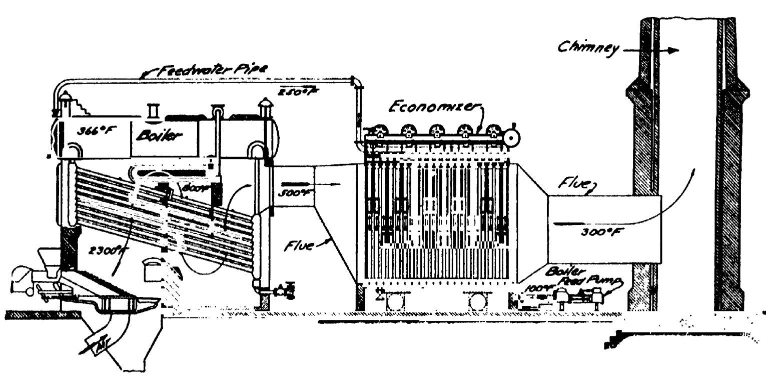

FIG. 155. — Section Drawing of Economizer Installation.

of a typical economizer installation is shown in Fig. 155, together with the temperature change and arrangement of feedwater piping.

Economizer elements are sometimes installed in the rear of the boiler setting and made a part of the boiler.

Ordinarily, the pressure in an economizer is equal to or slightly greater than that in the boiler. A type of multi-stage centrifugal turbine-driven feed pump is sometimes used, from the first stage of which the water passes to the economizer at low pressure, and returns to the remaining stages of the pump to receive an increase in pressure sufficient to enter the boiler. Old economizers may thus be kept in use longer, and, in general, the life of the economizer is lengthened.

The economizer is of particular value in stations carrying a periodic or regular overload, because it practically increases the heating surface and saves the heat that would otherwise pass to the stack at such times. The storage of hot water for a sudden demand is an additional advantage. Reduction in draft caused by the economizer depends upon the length of the economizer and arrangement of tubes. Mechanical draft is generally used.

The economizer serves as a feedwater purifier, like any closed water-tube heater, and also reduces, by a small amount, the soot otherwise discharged with the gases. The gain in overall efficiency of the plant, resulting from the use of an economizer, may vary from 3 or 4 per cent to 18 or 20 per cent, depending on the conditions. In some large central stations of recent construction, economizers have been omitted because the saving to be obtained was considered too small, under the conditions of operation in the plant, to justify their use.

258. Boiler Feedwater. — Water obtained from natural sources, such as rain water, water from creeks, rivers, lakes, wells, and the sea, is never free from impurity, the relative degree of purity being in the order named. Falling rain absorbs carbon dioxide, air and other gases, and in the earth becomes impregnated with mineral compounds, loaded with organic matter and mud in suspension, and often defiled by acid wastes from industrial plants.

Water suitable for boiler feed should not contain corrosive substances, and should be soft, that is, free or nearly free from mineral salts and impurities that make a hard coating, or scale. These salts cause "**temporary**" or "**permanent**" hardness, according to their nature. *The degree of hardness is generally stated in grains of solids per gallon;* for soft water, 1 to 10 grains; for moderately hard water, 10 to 20 grains; for very hard water, above 25 grains. Temporary hardness may be removed by boiling, and permanent hardness by chemical treatment of the water.

Boiler scale *is a solid deposit, or incrustation, caused by the action of heat and the concentration of impurities held in solution and suspension in the water.* The concentration causes crystallization of the salt out of the solution and deposits on the metal. *The principal scale-making materials are the sulphates, nitrates, and chlorides of lime and magnesium, which make a hard scale, and the carbonates of lime and magnesium, which*

make a softer scale. A boiler tube in which scale has been allowed to accumulate and also the appearance of a typical scale formation, are shown in Fig. 156.

259. Effect of Scale. — The presence of scale on boiler plate prevents the rapid transmission of heat to the water, and may cause the overheating

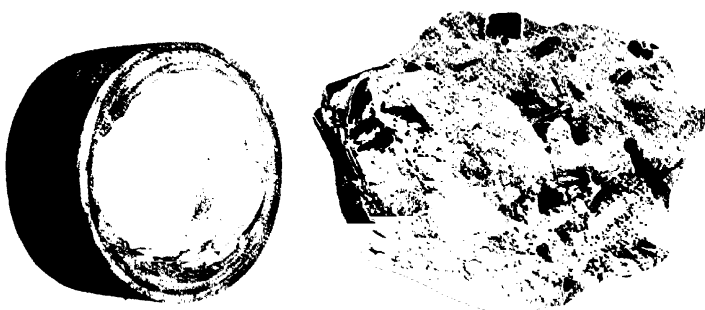

FIG. 156. — Boiler Tube showing Scale and a Typical Scale Formation.

of tubes, flues, or shell, with disastrous effects. Reports of boiler insurance companies show that a large proportion of the damage caused each year in steam boilers is caused by boiler scale.

Scale causes a loss in efficiency, because the gases pass away without giving up available heat. Tests made by Professor Schmidt at the University of Illinois, have shown that with hard and soft scale, varying in thickness from one-fiftieth to one-ninth of an inch, the loss of efficiency of heat transfer varied from 9 to 16 per cent of the heat supplied.

The losses resulting from the introduction of poor feedwater into the boiler are:

1. Waste of fuel on account of reduced efficiency, and also by taking boilers out of commission and heating them up after cleaning.
2. Labor for cleaning heaters and boilers.
3. Extra boilers required to allow for rotation in cleaning.
4. Cost of repairs, caused by overheating or corrosion of tubes and shell.
5. Cost of cleaning apparatus, such as tube cleaners.
6. Cost of boiler compounds.

260. Corrosion. — Corrosion, or wasting away, is caused by **electrolytic action** of metal in contact with water, and is especially active in the presence of acid and dissolved oxygen. *If the oxygen can be eliminated from the water, corrosion may be practically prevented.* The presence of

carbon dioxide, since it causes acidity, always implies a tendency toward corrosion; but if oxygen is not present also, the possibility of corrosion is slight. **Pitting** is due to local corrosion as the result of galvanic action between impurities, mill scale, and rust at points on the surface of the metal.

A notable example of the effect of corrosion is given in an article presented at the annual meeting of the American Society of Mechanical Engineers, December, 1915. In twelve water-tube boilers at the Frontvale plant of the Southern Pacific Company, one-third of the tubes required replacing in eighteen months after the plant had been placed in operation.

Slabs of zinc securely fastened to the stays of marine boilers are used to prevent corrosion of the boiler metal. The zinc requires renewal, as it is gradually wasted by galvanic action.

W. H. Walker, Ph. D., formerly Director of Industrial Coöperation and Research at Massachusetts Institute of Technology, says:

" In order to form rust, the iron must actually go into solution in the water; if oxygen is present it unites later with this dissolved iron, separating as red rust. When iron dissolves in water, an equivalent or corresponding amount of hydrogen is set free. This hydrogen covers the surface of the iron and protects it from further attack. But if oxygen be present it unites with the hydrogen, destroys the protective covering and allows the solution of iron to proceed."

The amount of corrosion has been found to be directly proportional to the free oxygen content of the water. To remove oxygen from the water as it enters a hot-water system, either a **deactivator** or a **deaërator** may be used. The deactivator removes only oxygen using a chemical process. It consists of a vessel containing many sheets of iron chemically treated so that rapid corrosion takes place in the presence of oxygen. The oxygen is thus consumed in corroding these cheap plates instead of the expensive piping system. This process has been used successfully on a large scale, and a description is given in the Transactions of the American Society of Heating and Ventilating Engineers, 1918, in a paper on " The Preservation of Hot Water Supply Pipe in Theory and Practice," by Apeller and Knowland.

The deaërator removes oxygen and all dissolved gases by a physical process. In one type of deaërator the water, from which the gases are to be removed, is sprayed into a vacuum chamber in which the vacuum is maintained at about 21 inches. The entering water is made to pass through a perforated brass plate and then it falls upon tubes heated by exhaust steam. The water is thus boiled violently at the temperature corresponding to the vacuum. The gases are released by the boiling and rise through a hole in the top of the brass plate to an ejector connection

located at the top of the deaërator. The Cochrane type of this apparatus combines the deaërator with an open feedwater heater operating under a vacuum that will give the desired temperature of feedwater. When thus made, the apparatus is known as a **vacuum deaërating heater.**

261. Methods of Scale Prevention and Water Softening. — The principal methods used to remove or neutralize the scale-making material in feedwater are:

1. Filters.
2. Boiler compounds.
3. Heating the feedwater.
4. Water softening .... { Intermittent system; Continuous system; Hot process; Zeolite process.
5. Evaporators.

The method to be used depends upon the amount and quality of the feedwater. Filters, which may be of either the pressure or gravity type, using granulated quartz or other sand as a bed through which the water is passed, may be used to remove material carried in mechanical suspension by the feedwater. In small installations having a fair quality of feedwater, the use of a boiler compound directly in the boiler is common. If there is a feedwater heater, a large part of the carbonates of lime and magnesium that make soft scale is precipitated out of the water by the heat, and may be removed in a manner depending on the construction of the heater.

262. Boiler Compounds. — Boiler compounds usually consist of carbonate of soda, to which caustic soda and phosphate of soda are sometimes added. Tannin and starchy materials are occasionally added, with the purpose of coating the particles of incrusting materials, holding them in suspension to prevent the formation of a solid mass. The boiler must be blown down periodically to prevent the accumulation of sludge, and occasionally opened and washed out thoroughly. The amount of compound to be used depends on the quantity of water evaporated, as well as its quality. The compound is sometimes pumped in continuously with the feedwater by a small pump made especially for that purpose, or forced in periodically by an injector or by use of a by-pass in the feed line. There is a tendency to look upon this method of treatment as a "**cure-all**" for any water difficulties This is a mistake, but when proper care is taken to suit the compound to the water in use, the results secured are fairly effective in the prevention of scale.

The **Navy Standard Boiler Compound** has been used successfully in a district having one of the worst waters used by locomotives on the Southern Pacific Lines and under conditions in which other treatment

was unsuccessful. This compound is composed principally of sodium carbonate but also contains tri-sodium phosphate, dextrine and a tannin compound. The tannic acid and starch are added to prevent the formation of scale, the action being to hold the impurities in suspension in a **colloidal,** or jelly-like, state. The tri-sodium phosphate prevents the rise of the surface tension of the solution and consequent priming caused by the impurities in the water and by the application of the other ingredients in the compound. In using this compound, a sufficient quantity must be added to each boiler to render the alkaline strength of the water in the boiler 3 per cent of the normal or above.

Kerosene and **graphite** are sometimes used in small quantities to loosen scale. They should be used with caution, especially where there is a chance for dislodged scale to accumulate and cause overheating of the metal.

A non-volatile mineral liquid, called "**perolin,**" injected into the boiler, enters through the scale to the surface of the boiler plate, and by expansion loosens scale already formed, or by coating the surface of the metal prevents the deposit of scale-making material.

263. Analysis of Feedwater. — The only satisfactory way to deal with the feedwater problem in taking water from an untried source is to begin with an analysis of the water. Companies which manufacture heaters, water-treatment apparatus, and boiler compounds maintain departments for that purpose; or the work may be done by a recognized chemist having suitable experience and equipment. It often happens that a water which appears to be suitable, on a casual examination, may carry corrosive elements, such as acid from mill waste, or bad scale-making material. The analysis of a well water and a river water are given in Table 23, which shows the incrusting or scale-making constituents in grains per gallon.

264. Effect of Heating Feedwater. — The result of heat alone on feedwater is shown by the column adjoining the analysis of the raw water in Table 23. The water was passed successively through an open heater and a live-steam purifier. It is evident that, when purification by heat alone is used, the larger the percentage of incrusting solids, the larger is the percentage of removal effected. In both cases, the diluting effect of the condensed exhaust steam was present in the second analysis, possibly amounting to over 25 per cent, or more than one-quarter, of the reduction noted in the solids. Time is an important factor in the extent of purification in the heater; to remove *temporary hardness* by boiling at atmospheric pressure, a period of about thirty minutes is required for precipitation of the carbonates of lime and magnesia to the limit of solubility. As the feedwater does not generally remain as long as this in the heater, the full effect is not realized.

TABLE 23. — ANALYSES OF A BOILER FEEDWATER

Name of Substance	Well Water Grains per U. S. Gallon		River Water Grains per U. S. Gallon	
	Raw	After Heating	Raw	After Heating
Calcium carbonate	18.39	2.50	3.51	1.26
Calcium sulphate	....	0.98	0.54	0.42
Magnesium nitrate	0.89	0.27	....	....
Magnesium sulphate	7.80	2.51	2.52	1.90
Magnesium chloride	2.13	1.13	....	....
Magnesium carbonate	1.97	....	....	....
Silica	0.85	0.15	0.75	0.60
Iron oxide and alumina	....	....	0.50	0.25
Sodium chloride	0.33	0.08	0.81	0.68
Volatile and organic	1.55	1.10	2.73	3.23
Total solids	33.91	8.52	11.36	8.34
Suspended matter	0.15	0.35	....	....
Free carbonic acid	0.99	0.06	1.00	....
Incrusting solids	33.58	8.44	10.55	7.66
Non-incrusting solids	0.33	0.08	0.81	0.68

265. Water-softening Apparatus. — Where considerable quantities of water are required for boiler purposes, and the supply is poor in quality, water-softening plants have come into use. The method of treatment is both **mechanical** and **chemical,** and the process may be **intermittent** or **continuous.** In the Scaife intermittent system of water purification, the apparatus, Fig. 157, consists of two or more **treating** and **settling tanks,** a **chemical mixing tank** with pump or injector, a **filter tank,** and a motor to drive the **mechanical stirrers** in the treatment tanks. While one treatment tank is being filled, treated and settled, the other tank is supplying treated water.

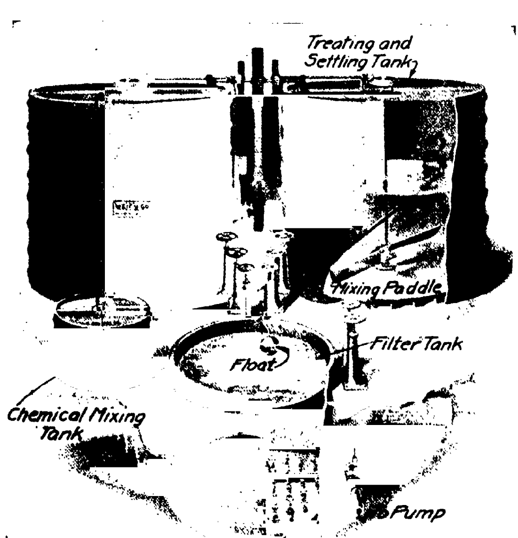

Fig. 157. — We-Fu-Go Water Softening System.

The exact quantities of the reagents, **lime** and **soda ash,** are weighed

to correspond to the quantity of water to be treated, and are mixed with water in the chemical mixing tank. The mixture is then pumped into the tank under treatment. The lime unites with the bi-carbonates of lime and magnesium, which produce temporary hardness in the water, and forms the insoluble carbonates of lime and hydrate of magnesium. These appear as a sludge at the bottom of the tank. The action of the paddles mixes the reagents and stirs up the sludge which has been allowed to remain from a preceding purification.

The sludge assists in settling the newly formed precipitate. After the mixing has stopped and sufficient time has been allowed for the precipitate to settle, the softened water is drawn out through a floating outlet pipe, from a point near the surface where it is cleanest. It passes to the filter tank, where the water level is controlled by a float and valve, and then to the supply tank for the boiler feed pumps, or to an open feedwater heater.

In the continuous type of apparatus, the water passes through compartments in which the reagents are continuously added during the stirring process, and then to a settling compartment and through a sand filter to the boiler feed line. In the "hot" continuous process, the action takes place more rapidly than in the cold process, and less storage capacity is required.

The chemical reaction equations for the removal of temporary hardness, together with the names of the components that enter and result from the combination, are as follows:

$$Ca(HCO_3)_2 + Ca(OH)_2 = 2\,CaCO_3 + 2\,H_2O$$
Calcium bi-carbonate + lime hydrate = calcium carbonate + water

$$Mg(HCO_3)_2 + CaO = MgCO_3 + CaCO_3 + H_2O$$
Magnesium bi-carbonate + lime = magnesium carbonate + calcium carbonate + water

$$MgCO_3 + Ca(OH)_2 = Mg(OH)_2 + CaCO_3$$
Magnesium carbonate + lime hydrate = magnesium hydrate + calcium carbonate

Magnesium chloride is also removed, and the resulting reaction is:

$$MgCl_2 + Ca(OH)_2 = Mg(OH)_2 + CaCl_2$$
Magnesium chloride + lime hydrate = magnesium hydrate + calcium chloride

The sulphates, nitrates and chlorides of lime and magnesia cause *permanent hardness* and dense scale in boilers. The soda ash, or sodium carbonate, precipitates insoluble calcium carbonate and also forms sodium sulphate. It also acts on the calcium chloride and makes calcium carbonate and sodium chloride. The sulphate and chloride of sodium are soluble. The reaction equations are as follows:

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4$$
Calcium sulphate + soda ash = calcium carbonate + sodium sulphate

$$CaCl_2 + Na_2CO_3 = CaCO_3 + 2\,NaCl$$
Calcium chloride + soda ash = calcium carbonate + sodium chloride.

266. Zeolite Process. — The reagent used for softening the water is an artificial material, **permutit,** made up largely of sodium compounds. The calcium and magnesium compounds in the water, being forced through a cylinder containing this material, unite with the sodium compounds and come out with **zero hardness.** The material has to be regenerated by means of a salt solution, which is passed through it. If the water entering the boiler has a large proportion of carbonate of lime, the large amount of sodium replacing the lime may cause foaming; in such a case all the water is first put through a continuous tank in which it is subjected to the lime treatment. It then goes to a filter and to the zeolite softener. The zeolite process is suitable for removing permanent hardness.

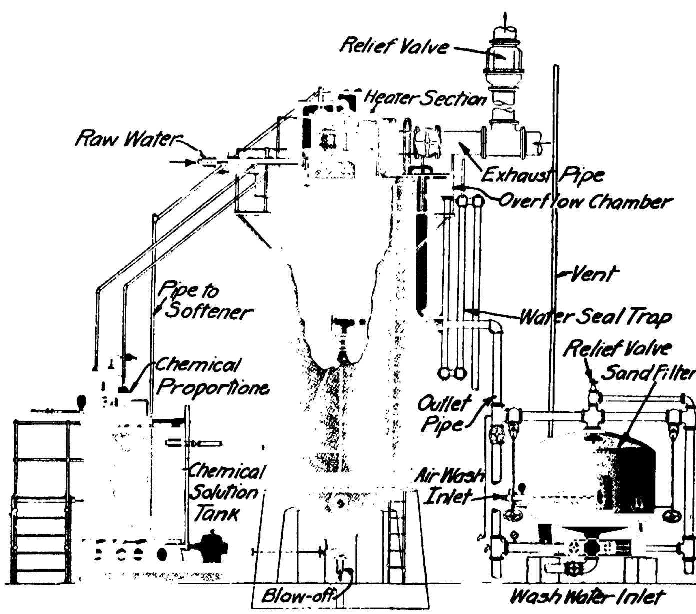

Fig. 158. — Sorge Cochrane Hot Process Water Softening System.

267. Hot-process Water-softener. — The Sorge-Cochrane water-softener supplies a chemical treatment to feedwater that has been heated to 205 deg. fahr. or higher. The effect of heat upon chemical reactions is to hasten them; and the rapid deposit of scale-making material is favored by increasing the temperature of the water, since this material descends more quickly in a medium of less density and viscosity. Since less of the scale material is held in solution in hot water than in cold water, less will pass along to the boiler.

This water softener, Fig. 158, consists of an open feedwater heater and oil-separator mounted above a tank in which chemical treatment and sedimentation take place. The hot water, freed from carbon dioxide, air, and other gases, is mixed with reagents from the automatic chemical proportioner, and falls into the tank below, where the scale material is deposited. The hot, treated water passes up into an inverted funnel, out to a sand filter, and thence to the feed pump and boilers. The sludge is blown out of the sedimentation tank by opening the valve at the bottom of the cone. The process is continuous. Where a part of the supply to the heater, such as condensate, does not require chemical treatment, a compartment is provided in the heater from which the heated water is supplied directly to the feed pump.

The advantages of water-softening in railroad work are as follows:

1. Reduction of fuel cost by 15 to 20 per cent.
2. Increased life of flues and fireboxes.
3. Increased road service of locomotives.
4. Less interference with traffic by failure of engines on the road.

The Union Pacific Railway now has thirty-seven plants in operation, having a total capacity of 417,000 gallons per hour, and removing over 3 tons of scale-making material per day.

268. Foaming and Priming. — Foam in boilers consists of unbroken bubbles of steam. It is caused by impurities which are dissolved in the water, or carried in suspension, and thus render difficult the free escape of the steam bubbles as they rise to the surface of the water. As the impurities become more concentrated, foaming increases. Scum, which may be caused by oil, vegetable matter, or sewage collecting on the surface of the water, is a frequent cause of foaming, because the bubbles of steam are prevented from breaking as they arise to the surface. If the water contains an alkali, it will change any vegetable or animal oil mixed with it into soap, which forms suds and causes foaming. Excessive foaming causes water to be carried into the steam pipe in sufficient quantities to become dangerous, because it is carried along by the rapidly moving steam and may rupture the piping. Foaming is generally attributed to sodium and potassium salts. It may be controlled by blowing down a part of the boiler contents and admitting fresh feedwater, and may be prevented by using suitable feedwater.

Priming consists of water carried from the boiler by the steam. It may be in the form of a mist, or, if the boiler is priming badly, the water may leave the boiler in belches. It is due to several causes: the rapid disengagement of steam from too small an area of water surface, the method of operation of boiler and engine, a high water level, overloading, and impurities in the water. PARSONS, in *Steam Boilers*, states that to avoid excessive priming the velocity of disengagement should not exceed 2.5 feet per second.

269. Evaporators for Distilled Make-up Water. — It has been well said that the only satisfactory compound to use in a boiler is pure feedwater. The use of distilled water practically eliminates the formation of scale and the attendant heat and mechanical losses, and allows boiler operation at large overloads without danger of foaming or priming. Evaporators for supplying distilled make-up feedwater have long been in marine use, and both high pressure and vacuum evaporators are now in successful operation in large stationary plants. The steam from the evaporators may be condensed in a feedwater heater by the condensate from the main condensers, or by the use of another condenser.

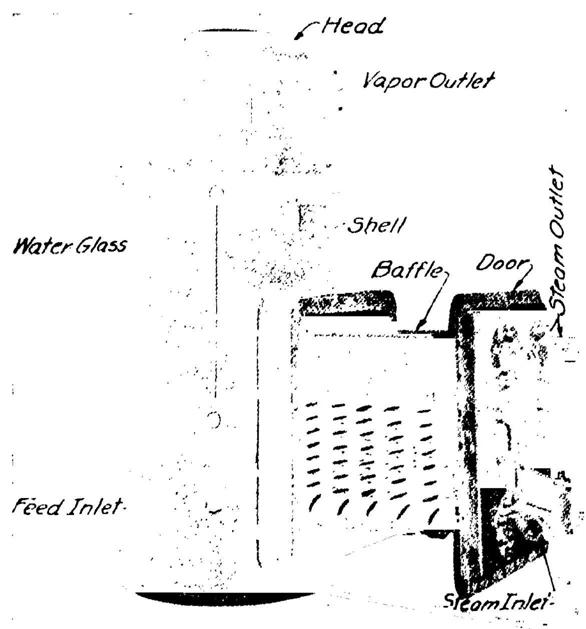

FIG. 159. — Shell Type Evaporator.

A typical shell evaporator is shown in Fig. 159. It consists of a closed cast-iron shell containing steam coils connecting top and bottom manifolds. Live steam at low pressure, or exhaust steam, enters the lower manifold and leaves the upper manifold. Removing the bolts around the door allows coils, manifolds, and door to be run out on rollers for cleaning and inspection. The lower manifold is drained to take care of the condensed steam. The evaporator is provided with water glasses to show height of the water, and also a safety valve. The locations of feed inlet, vapor outlet, and blow-off are marked on the illustration. A crown baffle is used to prevent water from being carried into the vapor outlet.

270. Scale Removal. — Although a considerable amount of scale-making material may be removed from the feedwater by using heaters, and by methods of water treatment, nevertheless, scale is deposited in boilers, and some mechanical means is required to remove it.

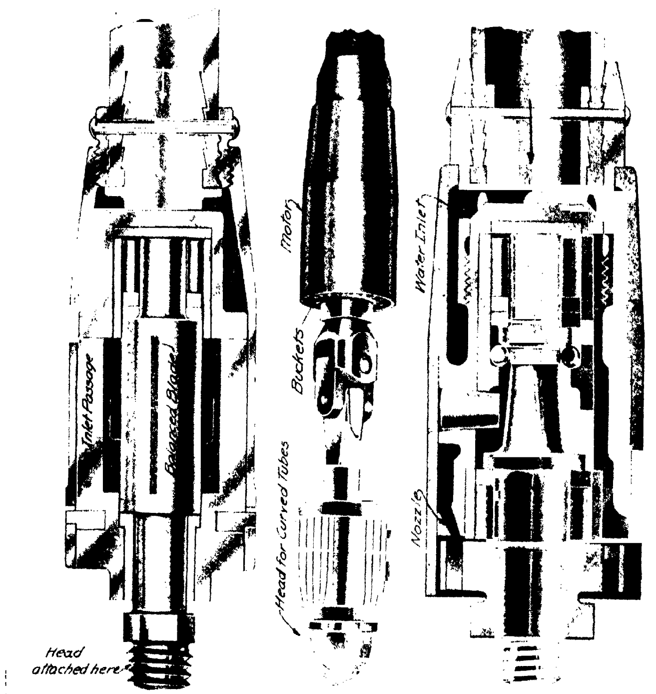

FIG. 160. — Pneumatic and Water-driven Tube Cleaners for Water-tubes.

In accessible parts of drums or shells, the scale is removed by using a hammer and a blunt chisel. For boiler tubes special cleaning devices are used; the tubes of water-tube boilers are freed from scale by passing through them a cleaner head carrying a tool rotated by a motor driven by air, steam, water, or electricity.

A **pneumatic cleaner,** using air at 50 to 75 pounds per square inch gage pressure, is shown in Fig. 160. Air is supplied through **armored hose** to the motor, which consists of a cylindrical casing in which revolves a shaft carrying the cleaning tool. The shaft is slotted parallel with its axis, to carry a sliding steel blade against which the air presses and thus revolves the shaft. Figure 161 shows the casing, blades, and inlet and exhaust ports. Oil for lubricating the motor is carried through the hose from a sight-feed oiler attached to the air-pressure pipe. Steam may be used in this cleaner, instead of air.

A tube-cleaner driven by a water turbine is also shown in Fig. 160. A pressure of 100 to 125 pounds per square inch gage is desirable for rapid work. The turbine wheel is at the open end of the hardened steel casing, and drives the rotating shaft which carries the cleaning tool. A steel shrouding band over the ends of the turbine buckets prevents loss of water radially and thus increases the efficiency.

The most important part of a mechanical cleaner is the **cutting head,** several types of which are shown in Fig. 162. The pins of the straight chilled-iron cutter wheels are centered in forged steel blocks that move freely in slots in the supporting disks. The head is often driven by a pulley, operated from a motor or engine by a belt, and the fragments of scale are carried away by water supplied to the boiler tube through a hose connection. When the motor is made a part of the cleaning head, air, water, or steam used for power is supplied through hose connections to the motor.

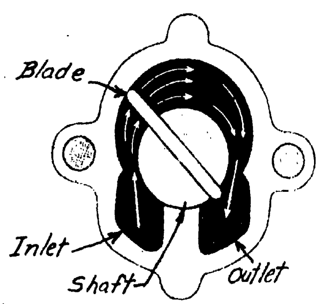

Fig. 161. — Section of Pneumatic Cleaner showing Operating Blade.

Electrically driven motors are located outside the tubes, and drive the cutter head by a straight shaft, or flexible jointed connections to allow for bends in the tubes.

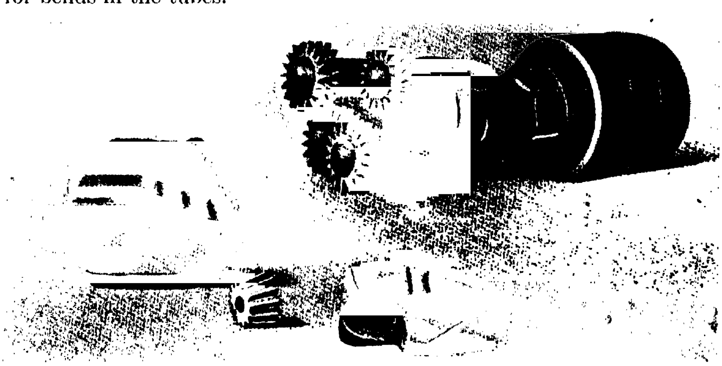

Fig. 162. — Cutting Heads for Scale Removers.

For cleaning scale from the outside of fire tubes, the cleaner head may be like those previously described, except that a swinging percussion tool or knocker head is attached to the rotating shaft; or the head may carry a vibrating plunger, Fig. 163. A large number of rapid, light blows are delivered against the inside of the tube, and the scale is cracked off as a result of the vibrations. Compressed air or steam should be used as the

motive power instead of water, in cleaning fire tubes, because it is generally inconvenient to dispose of the water in fire-tube boilers.

271. Soot Removal. — In small boiler plants, scrapers and brushes attached to rods made of small pipe are used by hand, for the ordinary cleaning of soot and light incrustation formed by particles of ash.

272. Soot Blower. — Several types of soot blowers, differing principally in the construction of the blower head, use steam for power. Some use

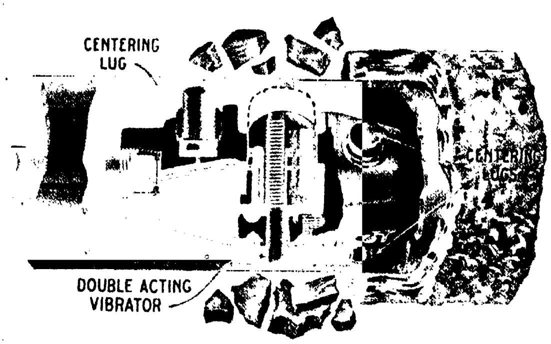

FIG. 163. — Vibrating Type of Fire-tube Scale Remover.

the steam blast only, while in others the steam jet induces a volume of air to assist in blowing out the soot. The latter form has greater cleaning power. In the efficient soot blower steam is expanded in a nozzle located in the blower head, Fig. 164, and attains a high velocity. On entering the boiler tube it draws in a large volume of air, and the combined blast

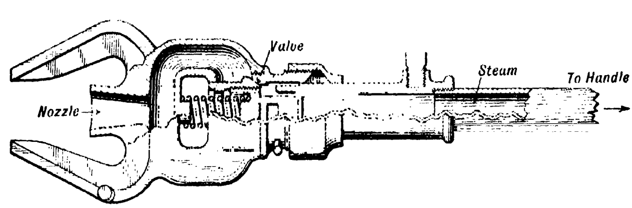

FIG. 164. — Hand-operated Soot Blower Head.

of steam and air sweeps the tube effectively. An open-ended steam pipe, called a **steam-lance,** is often used by hand to remove soot, but it is uneconomical in the use of steam.

A permanent installation of **steam jets,** Fig. 165, for the removal of soot, is becoming common in important plants, since the gain in economy more than warrants the expense. The mechanical device is more likely to be used frequently, as it is comparatively easy and agreeable to operate,

while cleaning the tubes with a hand blower is a disagreeable job likely to be neglected.

When applied to water-tube boilers the mechanical soot cleaner consists essentially of a cleaning element, which may be stationary or movable. A typical soot cleaner as applied to a B. and W. type boiler consists of one or more elements for each pass of the boiler setting. Each element is located above the tubes and at right angles to them, and has steam nozzles arranged to direct the steam, on the tubes, along diagonal paths at 60 degrees to the horizontal. Steam is supplied to the elements by pipes having hand controlled valves. Each element is mounted on bearings and is provided with a chain sprocket located outside the setting, so that the jets of steam can be made to sweep over the tubes and thus clean them more effectively.

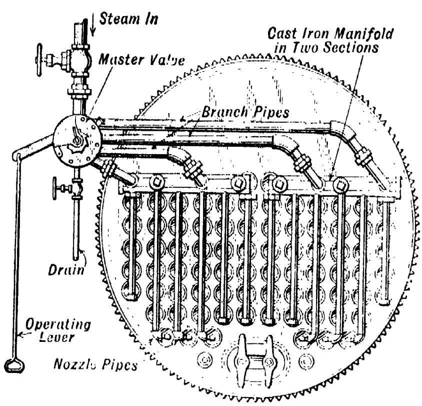

FIG. 165. — Stationary Nozzle Soot Blower attached to H. R. T. Boiler.

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REVIEW QUESTIONS

1. What is the primary purpose of heating boiler feedwater?

2. Name three methods of heating feedwater and explain the essential differences in the methods.

3. Describe the construction and operation of an open type of feedwater heater.

4. Describe the construction of an economizer. Where is the economizer located with respect to the boiler and chimney?

5. Name three methods of purifying feedwater.

6. Describe the construction and operation of an intermittent water-softening plant.

7. By what means is scale, which has accumulated on the surfaces of tubes, removed?

8. Describe the construction of the apparatus used to remove scale from a water-tube.

9. Describe the method of removing soot with a permanent soot-removal apparatus.

CHAPTER XIII

SUPERHEATERS

273. Foreword. — Modern power plants are using *superheated steam* to an increasing extent. As has been previously mentioned, superheated steam is steam which has been heated to a temperature above that corresponding to the pressure at which it was formed. Superheated steam effects a saving in fuel by:

1. Reducing condensation in the steam pipe lines.
2. Decreasing cylinder condensation in the engine cylinder, because of the poor conductivity of superheated steam.
3. Reducing the friction of the steam in the steam ports while entering and leaving the cylinder.
4. Producing more work per pound of steam used.

The poorer the economy of a steam engine, the greater is the saving from the use of superheated steam. The saving in the steam required

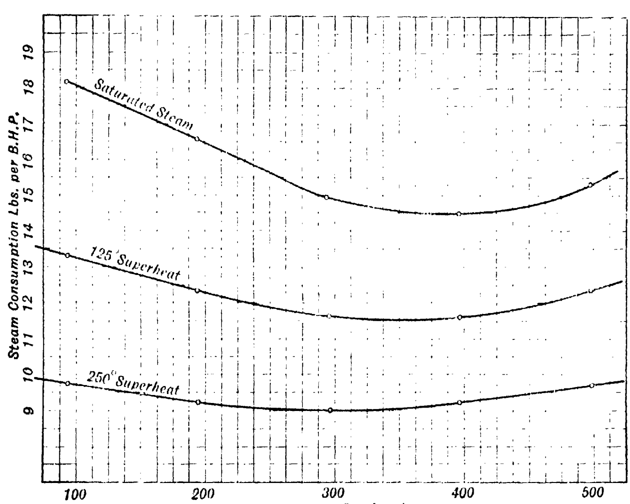

Fig. 166. — Steam Consumption of a 250-Hp. Compound Condensing Engine.

to produce a brake horsepower for different degrees of superheat is shown in Fig. 166 for a 250 horsepower compound engine, operating condensing.

In turbine practice superheated steam decreases corrosion of the blades caused by particles of water carried by the steam.

274. Methods of Superheating Steam. — Steam may be superheated in the shell of the boiler, as in the steam space surrounding the top of the fire tubes in large vertical boilers, or in special forms of superheaters of which the following types are in general use:

1. Attached superheaters.
2. Separately fired or direct superheaters.

The attached superheater forms a part of the boiler itself. It is located at some convenient point within the setting and in the path of the furnace gases. The direct-fired superheater is located in a setting of its own with an individual furnace to supply the heat required for superheating.

The **attached superheater** has four advantages: besides having (1) lower first cost, and (2) higher operating efficiency, it requires (3) less attention, and (4) less space. As ordinarily installed, it is subject to the fluctuating temperatures of the furnace, which produce a varying degree of superheat, depending upon the manner in which the furnace is being operated. Superheaters are sometimes constructed to maintain a constant degree of superheat automatically, but in most cases the superheat increases slightly with the load. The attached, or integral, type of superheater is most common in standard central station practice. It is used with both fire- and water-tube boilers, and with special construction is adapted to marine and locomotive boilers.

The **direct-fired superheater** has the following advantages:

1. Degree of superheat may be varied independently of the load on the boiler.
2. May be placed at any convenient point.
3. Repairs can be easily made without shutting down the boiler.

Its disadvantages are as follows:

1. Requires separate attention.
2. Extra space and piping are required.

275. Foster Attached Superheater. — The Foster superheater, Fig. 167, has an inlet and outlet header, into which are expanded a series of steel pipes through which the steam passes to be superheated. In order to present as large an area as possible to the heat, a closed steel tube is placed inside the straight portion of each pipe, and an annular space is thus formed for the passage of the steam. A series of cast-iron rings are shrunk over the outside of the steel pipes, to protect them from overheating when putting the boiler in service or when there is not a rapid movement of steam through the superheater.

The Foster superheater is shown in Fig. 168 attached to a return-tubular boiler and located in the rear pass of the setting. Steam leaves the boiler through the boiler nozzle and enters the inlet pipe of the super-

heater; as it passes through the superheater, its temperature is raised above that existing in the boiler. The final temperature attained depends upon the volume and temperature of the steam passing through the super-

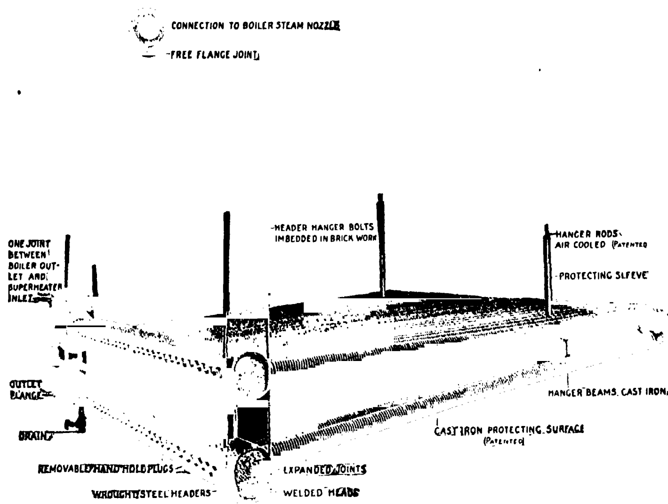

FIG. 167. — Foster Attached Type Superheater.

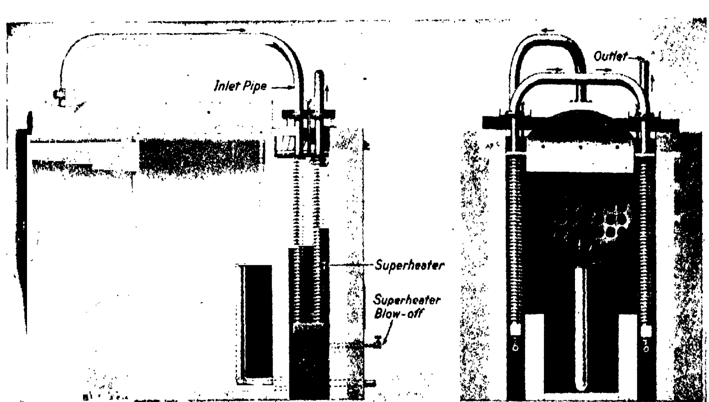

FIG. 168. — Foster Superheater on H. R. T. Boiler.

heater tubes, and upon the volume and temperature of the gases flowing over the tubes. In American practice the final temperature is seldom above 600 deg. fahr., while in European practice the final temperature is sometimes nearly 800 deg. fahr.

276. Babcock and Wilcox Attached Superheater. — This superheater has a construction similar to the Foster superheater, with the exception that the bent tubes are not surrounded by cast-iron protecting ferrules.' It is hung from the drums of the boiler by steel rods attached to angle irons on which the superheater rests, and provision is made to fill the

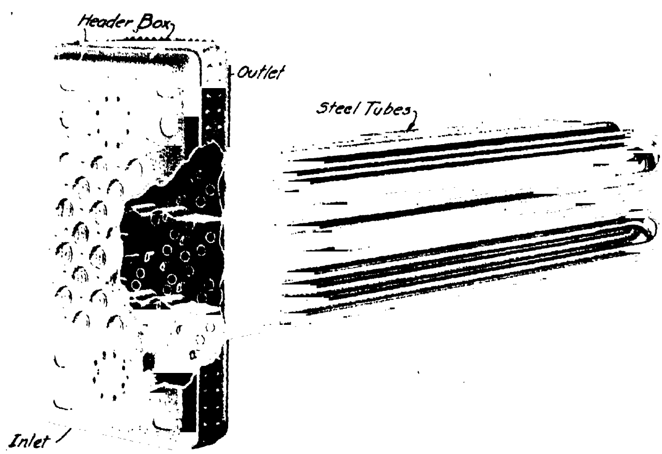

FIG. 169. — Heine Attached Superheater.

superheater tubes with water and thus prevent overheating when not in use.

277. Heine Attached Superheater. — The Heine superheater, Fig. 169, has a header, or box, similar to the Heine boiler-tube header, into which are expanded a group of U-shaped, seamless, drawn-steel tubes $1\frac{1}{2}$ inches in diameter. The flat surfaces of the box are braced by hollow staybolts, and handholes are provided in the front sheet of the box, for cleaning and repair purposes. Inside the superheater box are placed two partitions which separate the box into three chambers.

Steam enters the superheater box at the bottom, makes one pass through the lower tubes, returns through the second pass to the middle chamber of the box, whence it flows into the third pass and returns on the fourth pass to the upper chamber and the steam outlet.

The location of this superheater is in a special compartment of the setting, so arranged that gases from the furnace pass through a flue, located just back of the bridge wall, up around the superheater and into the last pass of the setting. A damper in the superheater chamber controls the volume of gases passing over the superheater, and may stop the flow of gases entirely. The superheater is supported by special castings and tile bars which rest on the brick setting.

278. Locomotive and Marine Attached Superheaters. — A typical locomotive superheater is shown in Fig. 170, located in the flues of a locomotive boiler. It consists of two headers and a number of small tubes

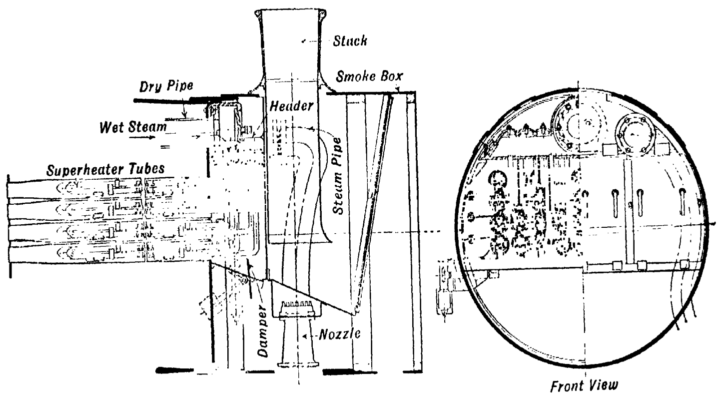

FIG. 170. — Locomotive Superheater.

connecting the headers. A Foster superheater is shown in Fig. 171, located in the second pass of a Foster marine boiler.

279. Direct-fired Superheater. — In cases where it is desired to have a greater control over the degree of superheat, the direct-fired superheater is used. It is similar to the attached type, but has a special furnace and setting, not connected with the boiler setting. By this method the steam from a number of boilers may be superheated by piping them all to the superheater.

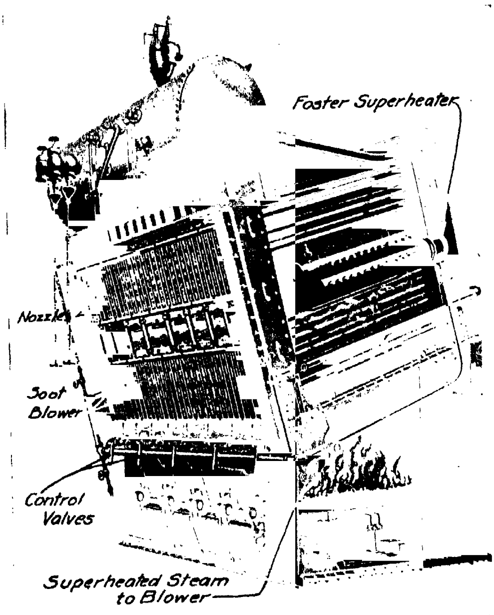

FIG. 171. — Foster Superheater and Diamond Soot Blower on Marine Boiler.

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REVIEW QUESTIONS AND PROBLEMS

1. State the advantages brought about by superheated steam.

2. Describe the construction of an attached type of superheater.

3. Find out to what extent locomotives operating in your neighborhood use superheated steam.

4. Referring to Fig. 166, find the saving in the amount of steam used by raising its temperature from that of saturated steam until it is superheated 150 deg. fahr. Is the percentage saving the same when the superheat is raised from 150 to 250 deg. fahr. as in the former case? What does this show regarding the point at which the greatest saving is made?

5. What is meant by the term direct-fired superheater?

CHAPTER XIV

DRAFT AND METHODS OF PRODUCING DRAFT

280. Foreword. — The steaming rate of a boiler depends upon the rate at which coal can be burned. This in turn depends upon the difference in pressure available to produce a flow of air through the fuel bed on the grate, to the chimney. This difference in pressure is known as **draft** and is required to overcome various obstacles which retard the flow of air. Draft is ordinarily measured by the use of a draft gage, and **its intensity is expressed as head in inches of water.**

The draft necessary for a given plant depends upon two factors: the pressure loss caused by retarding the gas flow by the breeching, boiler damper, baffle walls, and tubes; and the depth and kind of fuel on the grates.

The draft required to overcome the friction, caused by the flow of gases in the flue and breeching, is small. It amounts to approximately 0.13 of an inch of water per 100 foot length of flue and is increased 0.05 of an inch of water for each right-angle bend in the flue.

To overcome the resistance offered to the flow of gases by the baffle walls and boiler tubes, the amount of draft required varies with the rate at which the boiler is working, the type of boiler, and type of baffling used. Table 24 gives the draft required for a few typical boilers.

TABLE 24. — LOSS OF DRAFT IN BOILERS

Type of Boiler	Inches of Water
Horizontal return-tubular	0.25 to 0.30
Babcock and Wilcox	.20 to .35
Heine	.49
Stirling	.51
Wickes vertical tubular	.43
Cahall vertical tubular	.45

The pressure required to force the air needed for combustion through the fuel bed on the grate is one of the most important factors in determining the necessary draft. This pressure varies with the kind of fuel and the rate of burning, as shown in Table 25.

TABLE 25. — DRAFT BETWEEN FURNACE AND ASHPIT TO BURN COAL*

Kind of Coal	Coal Burned per Square Foot of Grate per Hour, Lb.						
	15	20	25	30	35	40	45
	Force of Draft Inches Water						
Bituminous, Ill., Ind., Kan	0.14	0.20	0.26	0.33	0.40	0.48	0.57
Bituminous, Ala., Ky., Pa., Tenn.	.16	.23	.31	.40	.49	.60	.72
Semi-bituminous................	.18	.26	.35	.45	.57	.71	.87
Anthracite Pea..................	.30	.45	.64	.88	1.23	...	...
Anthracite Buckwheat No. 1.....	.43	.68	1.00	1.50	...	...	...

* From Mechanical Equipment of Buildings, Vol. II, Harding and Willard.

The relative amounts of the various draft losses are shown in Fig. 172, together with the location of the draft gage connections for measuring the amount of the losses.

The addition of an economizer will increase the necessary draft by about

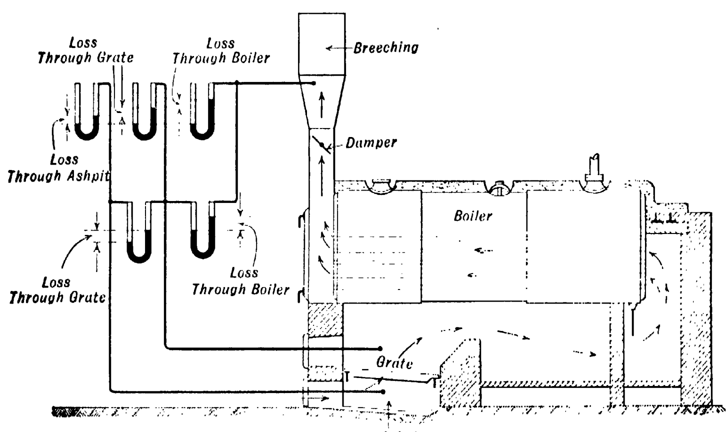


FIG. 172. — H. R. T. Boiler showing Location of Draft Gages to obtain the Various Draft Losses.

0.30 of an inch of water, and a superheater will add about 0.15 of an inch of water to the total amount of draft required without its use.

There are two types of draft: (1) natural draft, produced by chimneys; and (2) mechanical draft, produced by fans.

281. Natural Draft. — *Draft, as produced by a chimney, is caused by the difference in weight between a column of hot gases inside the chimney and the weight of a column of cold gases, of equal height, outside the chimney.* Chimney draft varies with climatic conditions, and is less when the

outside temperature is high. The most common types of chimneys are made of:

1. Brick.
2. Steel.
3. Concrete.

282. Common Brick Chimney. — A typical brick chimney, consisting of an outer and an inner wall with a space between, is shown in Fig. 173. The outer wall is made of well-burned hard red brick, and varies in thickness by a series of steps, from top to bottom, the minimum thickness at the top being the length of one brick. The **batter,** or taper, of the outer wall is approximately one-quarter of an inch per foot. The inner wall, or **lining,** is made of firebrick, and may extend through only a portion of the height of the chimney; it is better, however, to have it extend to within 8 or 12 inches from the top. The inside diameter of the lining is ordinarily uniform, with the offsets made on the outside. At the top of the chimney the outer wall is **corbeled** and drawn in over the opening between the walls. The top is then covered by a cast-iron or terra cotta **cap,** as shown in Fig. 173, to protect the top from the action of the weather and to hold it in position. The use of two walls prevents the unequal expansion of inside and outside portions of the relatively thick outer wall, and the loss of heat by radiation is to a small extent decreased. The chimney rests upon a substantial foundation, and a **clean-out door** and **flue connection** are provided near the base. Brick chimneys are generally made circular, although square, hexagonal, and octagonal sections are often used.

283. Radial Brick Chimney. — This type of chimney is circular in section and is made of moulded radial perforated brick, which are curved to conform to the inside and outside curvature of the chimney. The perforations aid in securing good quality of brick during burning and serve to increase the bonding action between bricks. The bricks vary in size and are laid as illustrated in Fig. 173, each row of brick being covered with a thin layer of cement. The small dead-air spaces, thus formed in each layer of brick, provide an excellent insulation and, by keeping the chimney gases hot, aid in producing good draft. The sides of radial brick chimneys usually taper 2 feet in 100 feet of height. An inner lining usually extends to a height of 35 or 50 feet above the flue connection.

A chimney of this type, located at the plant of the Boston and Montana Consolidated Copper and Silver Mining Co. of Great Falls, Montana, has a height of 506 feet above the foundation, and an inside diameter at the top of 50 feet.

284. Foundation for Brick Chimney. — A suitable foundation is necessary, to prevent the chimney from settling or tipping. It is usually constructed of concrete and is distributed over sufficient area to prevent set-

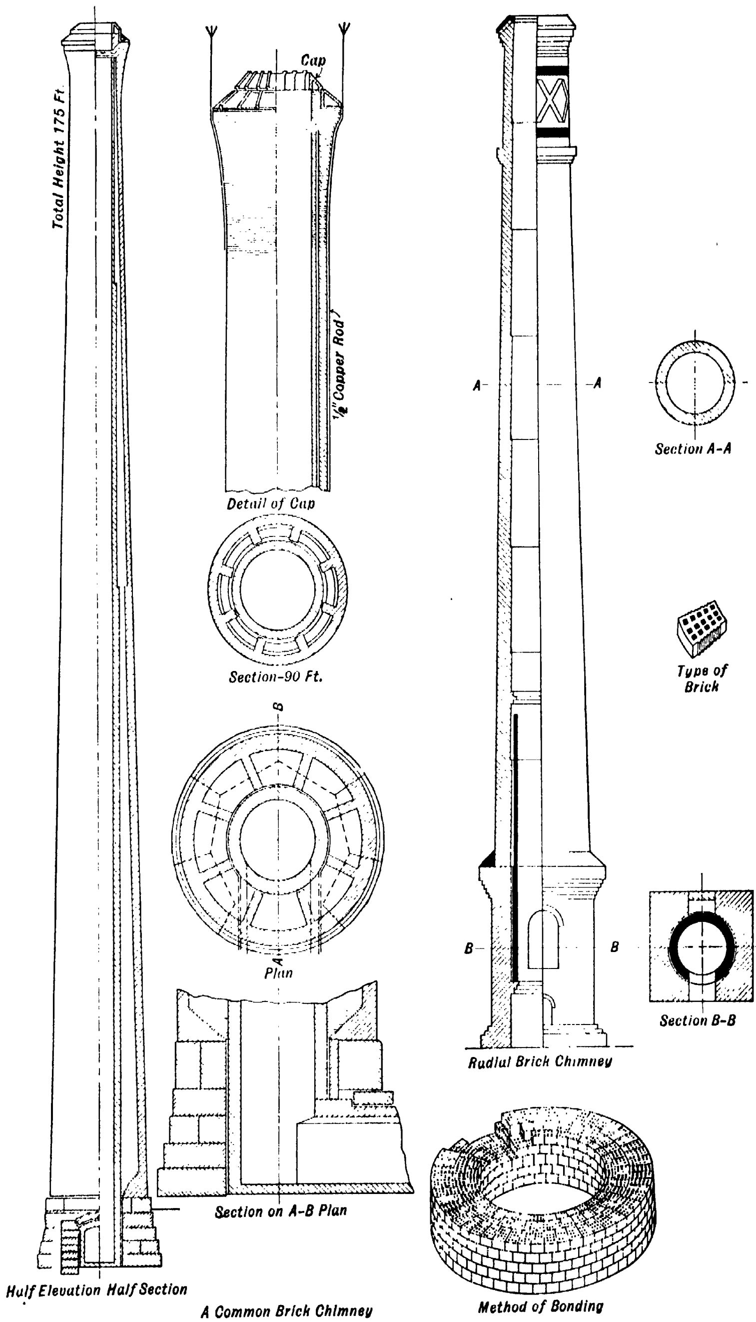


Fig. 173. — Brick Chimneys.

tling. The foundation of the chimney mentioned in the previous article is 111 feet square and $22\frac{1}{2}$ feet deep.

285. Steel Chimneys. — Steel chimneys, or stacks, are either self-supporting or guyed. The **self-supporting** steel chimney, shown in Fig. 174, is generally made of a series of rings having single-riveted lap joints. The steel plate varies from $\frac{3}{16}$ to $\frac{1}{2}$ inch in thickness, depending upon the size of the chimney. Cylindrical rings of two different diameters may be used in one chimney, the wider alternating with the narrower; or, all the rings may be of the same diameter and slightly conical, to allow the top of each ring to fit inside the bottom of the ring next above. The various rings are fastened together by a single row of rivets, and the bottom ring is flared to form a base which is held to the foundation by anchor bolts passing through lugs on the base. The steel structure above the boiler is often used to support the stack, in order to reduce the ground space required.

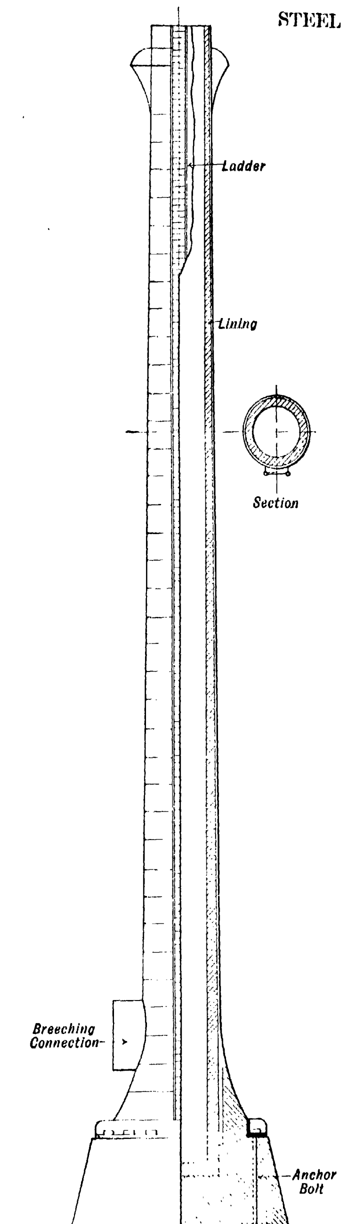


Fig. 174. — Self Supporting Steel Chimney.

This type of chimney depends for its stability on the anchor bolts and the foundation. It is ordinarily lined to prevent excessive radiation and to protect the inside from the corrosive action of the gases; the first 50 feet of the lining is made of firebrick and the remainder of common red brick. The lining is ordinarily of uniform thickness, set in contact with the steel and thoroughly grouted to prevent rapid depreciation. It is sometimes made in independent sections resting on a bracket riveted to the

shell. The stack is reinforced at the opening to which the breeching connection is attached.

The **guyed** steel chimney may be supported on a foundation, or may rest directly on the breeching. Such chimneys are short and limited in weight, since they are often supported directly by the boiler shell. The pressure of the wind is generally resisted by three steel cables, the upper ends of which are fastened to an angle or T-band attached at about two-thirds the height of the chimney, and the lower ends to the ground or building at such a distance from the chimney that the guys make an angle of about 60 degrees with the vertical.

286. Concrete Chimney. — The concrete chimney, Fig. 175, although used only in the last few years, is finding favor with many engineers, and its use is increasing. The sides of the chimney are made either straight or tapered. When they are straight, the thickness of the wall at the top is about 6 inches, and when tapered about 4 inches, with a taper of 4 inches per 100 feet. Since concrete is strong in compression and weak in tension, vertical steel rods are used to reinforce the concrete, in order to resist the tensile strains resulting from the pressure of the wind. Horizontal rings of steel are used to take up stresses caused by temperature changes, and to assist in holding the vertical rods. The walls are monolithic, and air leakage is thus prevented. Concrete chimneys should ordinarily be lined with hard-burned brick to about one-third of their height, and the space between the lining and the outside wall should be covered with a concrete cap.

287. Comparison of Chimneys. — The life of brick and concrete chimneys is longer than that of steel chimneys, the former retaining their usefulness for about fifty years, and the latter for five to fifteen years, depending upon the care taken to prevent corrosion. Because of cracks and imperfect bonding of the bricks, the brick chimney may have considerable air leakage, which the concrete and steel chimneys do not have.

The concrete chimney is light in weight, requires a smaller space than either steel or brick chimneys, has great resisting power, and can be rapidly constructed, 6 feet per day being an average rate. If unlined, it may have an excessive loss of heat by radiation, which tends to lower the draft produced.

The steel chimney can be constructed at a more rapid rate and has less weight than either the brick or the concrete chimney. It, however, requires frequent painting to prevent corrosion, which is rapid in air that contains salt water, as when near the ocean.

The cost of the brick chimney is the highest, and that of the unlined steel stack the lowest. The cost of concrete chimneys, based on costs in 1917, varied from $2,000 for a chimney having an inside diameter of 4 feet and a height of 105 feet, to $18,000 for a chimney with an inside

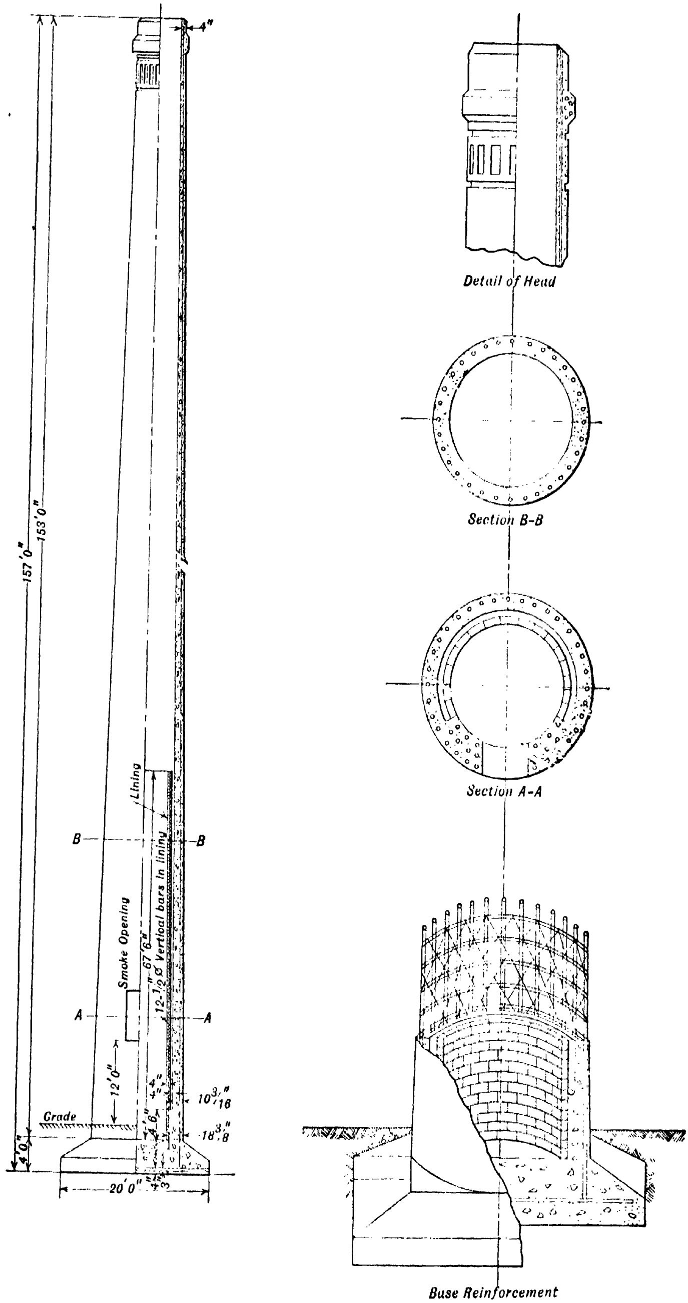


FIG. 175. — Weber "Coniform" Concrete Chimney.

diameter of 16 feet and a height of 258 feet. These values give some idea of how the cost of chimneys varies.

288. Chimney Draft Calculations. — The *height of a chimney* required to produce a certain draft is obtained by calculation, using the weight of gases inside and the weight of air outside the chimney. This height depends upon the kind of fuel used, because the draft required varies with the kind of fuel The *capacity of a chimney* is the number of cubic feet of gases the chimney will discharge per hour. It depends upon the area of the chimney and the velocity of the gases. The capacity determines the number of pounds of coal a plant will burn and it is influenced by the ratio of the height of the chimney to the diameter. In general, the diameter should be about 8 per cent of the height.

Consider a chimney filled with hot gases and closed at the bottom by a horizontal diaphragm. The pressure of the air, at the top of the chimney, produced by the atmosphere above that level, is the same on the gases inside and the air outside the chimney. *The pressure on the upper side of the diaphragm* is, therefore, the sum of the air pressure at the top of the chimney plus the pressure produced by the column of hot air inside the chimney. *The pressure on the under side of the diaphragm* is the pressure of air at the top of the chimney plus the pressure produced by a column of cold air equal to the height of the chimney. The air inside the chimney is at a higher temperature than the air outside, and consequently has less weight per cubic foot. The **draft** produced by the chimney, therefore, equals the difference in pressure on the two sides of the diaphragm and may be written as follows:

$$D = \frac{12\,H}{K}(d_c - d_h) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

in which D = intensity of draft in inches of water.

K = density of water in U-tube, pounds per cubic foot.
= 62.4 for a temperature of 70 deg. fahr.

H = the height of the chimney above the grate level, feet.

d_c = density of the cold air outside the chimney, pounds per cubic foot.

d_h = density of the hot gases inside the chimney, pounds per cubic foot.

With the diaphragm removed, as in an actual chimney, this difference in pressure produces a movement of gases from the bottom to the top of the chimney. When one leg of a U-tube partly filled with water is connected to the inside of the chimney and the other leg open to the air, the level of the water in the two legs will be displaced, as shown in Fig. 63, page 75. The difference in level between the surface of water in the two legs is the draft, measured in inches of water.

Equation (76) may be put in a more convenient form by substituting, for the values d_c and d_h, terms containing the corresponding absolute temperatures. This can be done by using the combined law of gases, $PV = WRT$, in which P is the absolute pressure in pounds per square foot; V = volume in cubic feet; W = weight in pounds; R = a constant = 53.37 for air; and T = absolute temperature, deg. fahr.

By writing $V = 1$, and P = pressure in pounds per square foot at sea level = 14.7 × 144 = 2116.8, in the above equation, W will equal the density, or weight per cubic foot, at atmospheric pressure; and for conditions outside and inside the chimney

$$W = d_c = \frac{39.7}{T_c}, \qquad \text{and} \qquad W = d_h = \frac{39.7}{T_h}$$

in which T_c and T_h equal respectively the absolute temperature of the air outside and the gases inside the chimney.

Substituting these values of d_c and d_h in Equation (76) and assuming the density of the gases to be the same as that of air at the same temperature, there results:

$$D = 7.64\,H\left(\frac{1}{T_c} - \frac{1}{T_h}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (77)$$

in which

D = the total draft produced by the chimney in inches of water.
H = the height of the chimney above the grate in feet.
T_c = absolute temperature of outside air, deg. fahr.
T_h = absolute temperature of inside gases, deg. fahr.

When the gas temperature is taken as it leaves the boiler, the draft available is taken as 0.80 of D in Equation (77), to allow for the cooling of the gases in the flue and stack.

Example 32. — A chimney 100 feet high is filled with gases at 600 deg. fahr., when the outside air temperature is 60 deg. fahr. Find the draft produced in inches of water.

Solution. — Using Equation (77) and making proper substitutions,

$$D = 7.64\,H\left(\frac{1}{T_c} - \frac{1}{T_h}\right) = 7.64 \times 100\left(\frac{1}{520} - \frac{1}{1060}\right) = 0.75 \text{ inches water.}$$

H = 100 ft.; T_h = 600 + 460 = 1060 deg. fahr.; T_c = 60 + 460 = 520 deg. fahr.

289. Empirical Chimney Formulae. — There are various empirical equations used to determine the area and height of chimneys. The most common of these equations is that proposed by William Kent, and is as follows:

$$\textit{Boiler horsepower} = 3.33\,E\sqrt{H} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (78)$$

in which E = effective area of a chimney in square feet. For circular chimneys, Kent considered a dead air space 2 inches wide extending around the inside of the chimney.
= $A - 0.60\sqrt{A}$, in which A = actual area of chimney, square feet.
H = height of chimney in feet.

Kent's equation assumes the burning of 5 pounds of coal per boiler horsepower per hour. For other rates of combustion Equation (78) should be multiplied by the ratio of 5 to the new rate of combustion.

Example 33. — Find the necessary area to discharge the gases formed in a boiler plant of 981 boiler horsepower burning coal at a rate of 5 pounds of coal per boiler horsepower. Height of chimney to be 200 ft.

Solution. — Using Equation (78)

$$E = \frac{\text{Boiler hp.}}{3.33\sqrt{H}} = \frac{981}{3.33\sqrt{200}} = 20.83 \text{ sq. ft.}$$

Boiler hp. = 981, H = 200 ft., E = effective area.

The effective area $20.83 = A - 0.6\sqrt{A}$

Solving for A the desired area is 23.76 square feet and the corresponding inside diameter 66 inches.

290. Mechanical Draft. — Draft, as produced by a chimney, depends upon the height of the chimney, temperature of the gases, and atmospheric conditions. In order to reduce the necessary height of chimney and at the same time obtain a draft that is independent of weather conditions and is easily and positively controlled, fans and blowers are used.

Two systems, known respectively as induced draft and forced draft, are used to produce draft by mechanical means. In the **induced draft system** the pressure over the fuel bed is reduced below that of the atmosphere, by means of a fan located between the boiler setting and chimney. Air passes through the fuel bed because of the difference in pressure above and below it. In the **forced draft system,** air is forced through the fuel bed, and the gases are removed from over the fuel bed by a chimney or induced draft fans. In each of these systems the power required to operate the fans varies from 4 to 6 per cent of the boiler capacity. In many power plants using underfeed stokers, induced draft is used to remove the gases from the furnace and forced draft to supply air, under pressure, for burning the fuel.

291. Induced Draft System. — A typical induced draft installation for stationary boilers is shown in Fig. 176. In this system the fans handle hot gases and, for a given weight of gases, must be of larger capacity, in the ratio of approximately 2 to 1, than a forced draft installation, on account of the increased volume of the hot gases. The fans are generally installed in duplicate and are located on a suitable foundation directly above the smoke connection. They are usually driven by a small vertical steam engine, which is automatically regulated by a balanced throttle valve in the steam line to the engine. This valve is under control of the boiler steam pressure and opens or closes to start or stop the fan and thus maintain the draft required to hold the pressure.

The draft for a locomotive is produced by using the exhaust steam from

the steam engine and air pumps. This steam is discharged from the steam cylinders through a steam nozzle, Fig. 177, to the stack. The form of the nozzle is such that the jet of exhaust steam has a conical shape and

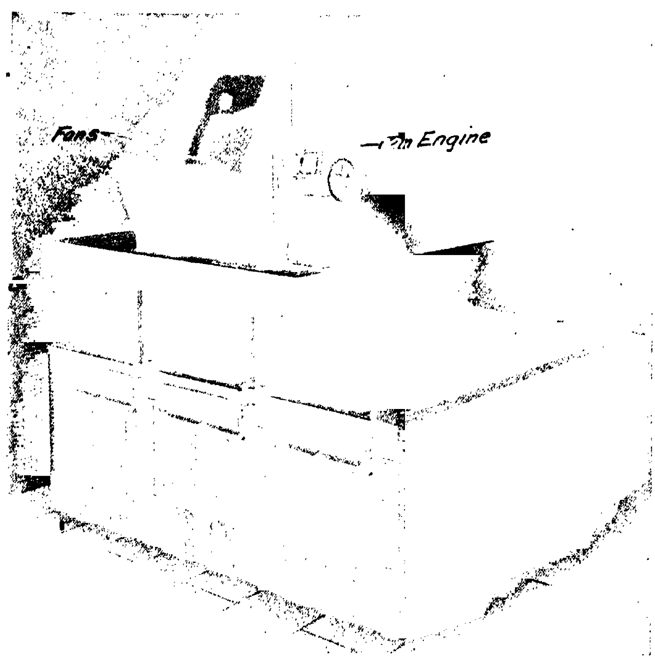


FIG. 176. — Induced Draft Installation.

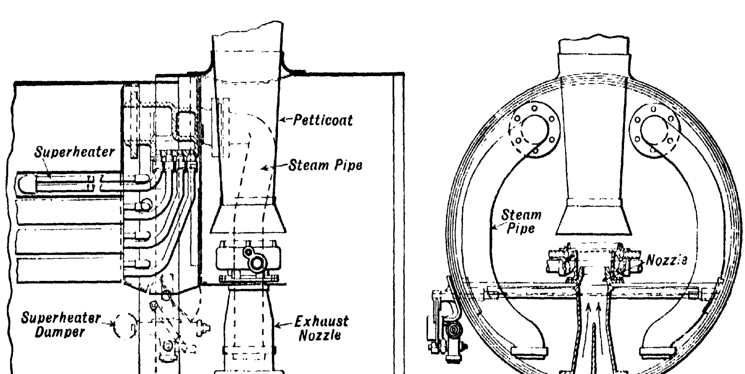


FIG. 177. — General Arrangement of Smokebox for a Pulverized Fuel Burning Locomotive.

completely fills the stack. A draft of about 12 inches of water is produced by this method. Because of the high draft, coal can be burned at the rate of 200 pounds per square foot of grate surface per hour, while, with chimney draft 25 pounds per square foot of grate surface per hour is considered high.

The **Prat, or Evasé, system** of producing induced draft is used extensively in Europe but has not been used to any extent in the United States.

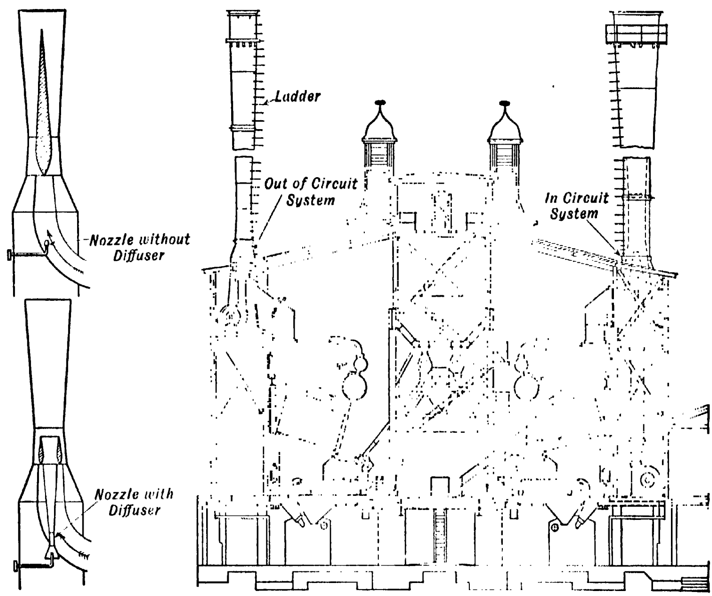


Fig. 178. — Prat or Evasé System of Induced Draft.

This system of draft uses a blower and a double tapered short stack, Fig. 178, which causes a draft on the principle of the ejector. Near the bottom of the stack, a section formed like an inverted frustum of a cone contains a nozzle through which the air from the fan is discharged. The next section above this is the throat, which is cylindrical in form; and from here to the top the stack flares outward and acts as a diffuser to diminish the energy loss, which would result from discharging the gases at high velocity into the air.

Two arrangements of the Prat system are used; in the first, the blower uses atmospheric air as the motive gas, while in the second it takes a small portion of the stack gases. For light loads the inspirator effect of the chimney is sufficient, but for heavier loads the blower forces air or gas at relatively high velocity into the throat of the stack and thus produces sufficient draft to handle the load.

292. Forced Draft System. — This system of draft is required by all underfeed stokers carrying a thick fuel bed, where the draft necessary

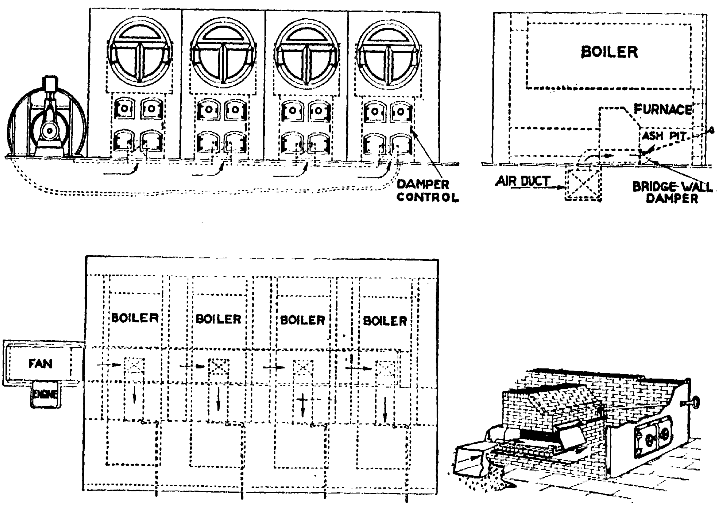


FIG. 179. — Method of Installing Forced Draft, for Hand-fired Furnace.

for satisfactory operation generally varies from 4 to 6 inches of water. As applied to stationary practice air is forced into a closed ashpit at the bridge wall as in Fig. 179, side wall, or bottom, and finds its way up through the fuel bed. A metal or concrete duct, which should be smooth with a minimum number of bends, connects the fan outlet with the opening into the ashpit.

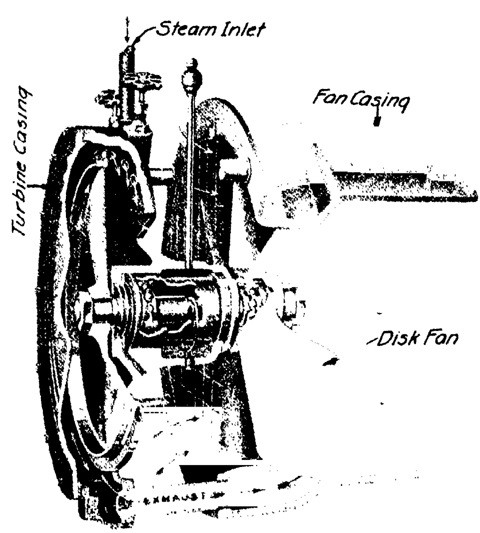


FIG. 180. — Coppus Turbine-driven Blower for producing Forced Draft.

A recent type of forced draft installa-

tion, which is used for both stationary and marine service, consists of a small turbine-driven blower, Fig. 180. When used for stationary service, the blower is placed at an opening in the side wall of each boiler setting; and when used for marine service is generally supported by a steel framework attached to the frame of the ship and above the firing room. This type of blower is well suited to the forced draft system, since it can be operated at high speeds and is easily controlled.

For marine work, two systems of forced draft are used, namely, the closed ashpit and the closed boiler room. In the **closed ashpit system,**

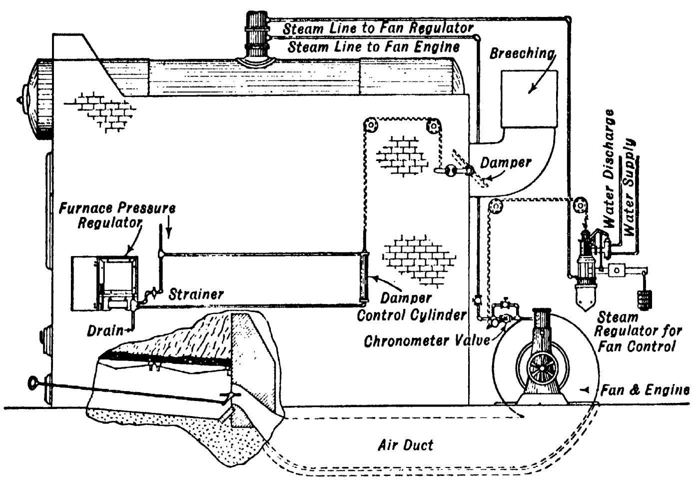


FIG. 181. — Diagram showing Balanced Draft with Steam Controlled Fan and Damper Regulator.

air is supplied by a suitable fan, through a duct, to the ashpit of the boiler, which is made air-tight. In this system the draft should be shut off when firing, to prevent forcing fire into the boiler room. With the **closed boiler room** the fan is located either in the boiler room compartment, as in Fig. 467, page 565, or at a suitable location outside the boiler room, and connected to the boiler room by suitable ducts as shown in Fig. 6, page 14. The boiler room is then made tight against air leakage and the ashpits are open. When this method is used, the boiler tubes are sometimes injured by a rush of cold air through the firing doors during firing. This difficulty is not encountered when oil burners are used.

293. Balanced Draft System. — As ordinarily used, the term "balanced draft" means a combination of forced and induced draft, so regulated that the pressure above the fire is maintained approximately atmospheric. Several companies manufacture equipment for automatically maintain-

ing the proper draft above the fire. A typical balanced draft system is shown in Fig. 181, where the various operating parts are named. With this system the draft over the fire can be maintained within 0.02 of an inch of water of that desired. The system operates as follows: When an increase of load occurs, the pressure will drop in the steam header; the steam regulator will then speed up the fan, thus forcing more air through the fuel bed and increasing the rate of combustion. This will produce more gas and tend to decrease the furnace draft. The furnace pressure regulator now increases the damper opening sufficiently to remove the gases at the new rate and thus maintain the pre-determined draft. With stoker installations, a compensating stoker can control the rate of fuel fed to meet the demand.

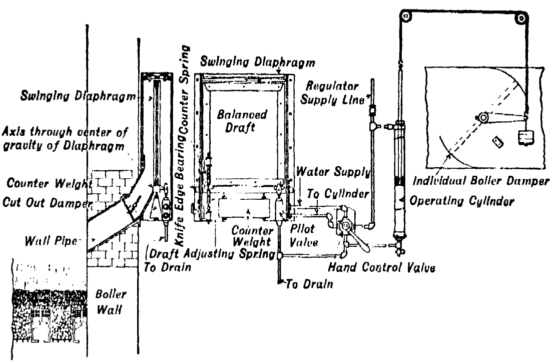


FIG. 182. — Balanced Draft Furnace Pressure Regulator.

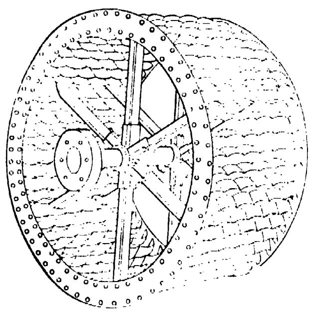
The **steam regulator,** Fig. 181, is a refined boiler damper regulator, which controls a reducing valve in the steam line to the forced draft fan and thereby controls its speed. The furnace pressure regulator, Fig. 182, is mounted on the side wall of the furnace and connected with the furnace, above the fuel bed, by a draft tube. A **swinging diaphragm,** mounted on **knife-edge bearings,** is drawn inward by the furnace suction and outward by a spring which is adjusted to maintain the desired draft. As the diaphragm moves it changes the position of the piston in the damper-operating cylinder and increases or diminishes the flue opening.

294. Fans Used to Produce Draft. — The most common types of fans used to produce draft are the following:

1. Steel plate and planoidal.
2. Multi-blade.
3. Conoidal.
4. Radial flow.
5. Centrifugal blower.

The construction of the first three types, Fig. 183, is essentially the same, with the exception of the impeller. Each of these types consists of a **rotating spider,** to which the plates or vanes are attached.

The spider is supported on a shaft, which rotates in bearings, generally ring-oiled and water-cooled when used for induced draft. The **casing** is made of sheet iron. The inlet is at the side of the casing and, as the **impeller** is revolved, air is delivered at the circumference of the impeller and flows along the casing to the outlet at a pressure equal to the centrifugal pressure generated by the revolving impeller. The multi-vane fan is the most efficient of these three types.



Multi-vane

The radial flow type of fan, Fig. 184, has an impeller so constructed that it causes a radial flow of the air. This makes it more efficient than the types previously considered.

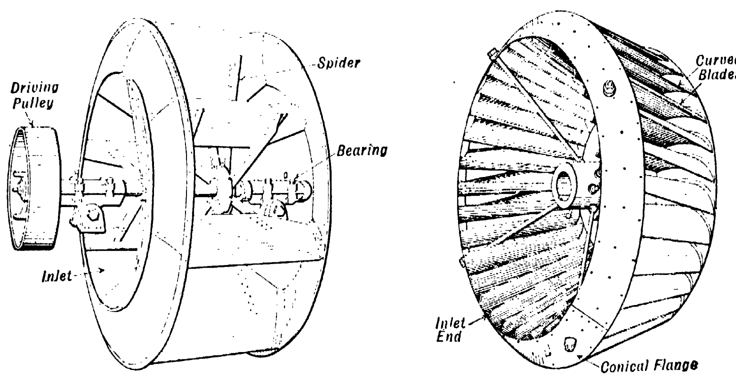


Plate *Conoidal*

FIG. 183. — Typical Fan Impellers.

295. Breeching Connections. — Adjacent to the stack and connecting the stack to the boilers, there is usually a smoke flue or breeching. This breeching may be circular or rectangular in section, and is made of steel plates, lined to reduce the loss of heat. Its area is generally 10 to 15 per cent greater than the area of the chimney, with the branch connections entering the flue in such a manner that the gas stream is disturbed as little as possible. Several forms of breechings are shown in Fig. 185.

296. Damper Regulator. — When natural draft is used, the amount of draft is regulated by changing the position of the damper in the uptake. To increase the efficiency and save fuel, the damper is commonly controlled by a damper regulator, which may be operated by water pressure, by steam pressure or by some type of thermostat.

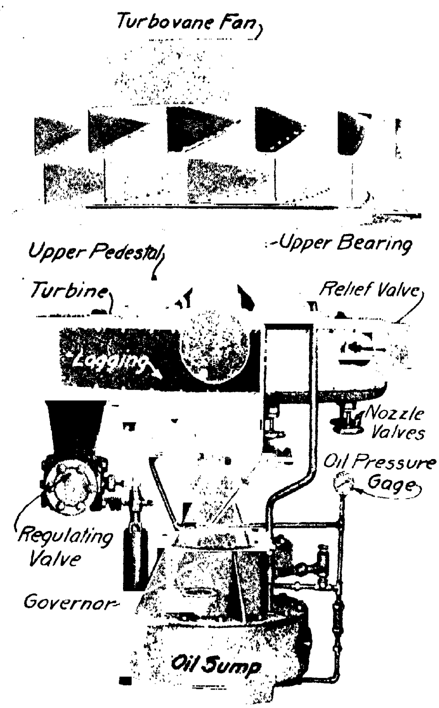


FIG. 184. — Turbo-vane Turbine-driven Fan.

A typical hydraulic type of damper regulator is shown in Fig. 186, in which the damper is normally held open by the counter-weight on the lever arm. This is the position for maximum draft. Steam pressure from the boiler acts on the upper side of the phosphor-bronze diaphragm in the steam chamber and is balanced by the counter-weight on the lever. The pilot valve is connected to this lever and controls the admission and exhaust of the water under pressure, to and from the power cylinder, which is proportioned to give a pressure seven times the water pressure. The power-cylinder piston rod is connected to the damper lever by a chain or rigid rod. With increase in steam pressure above that at which the regulator is set, the pilot valve

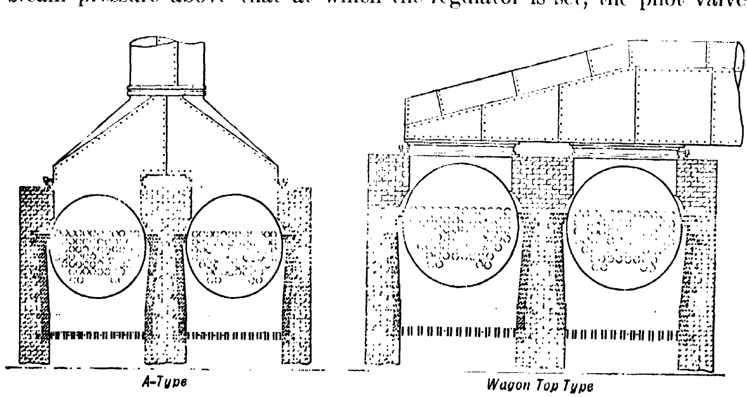


FIG. 185. — Typical Breeching Shapes.

is raised and water pressure admitted to the power cylinder, which partly closes the damper and decreases the draft. With a fall in pres-

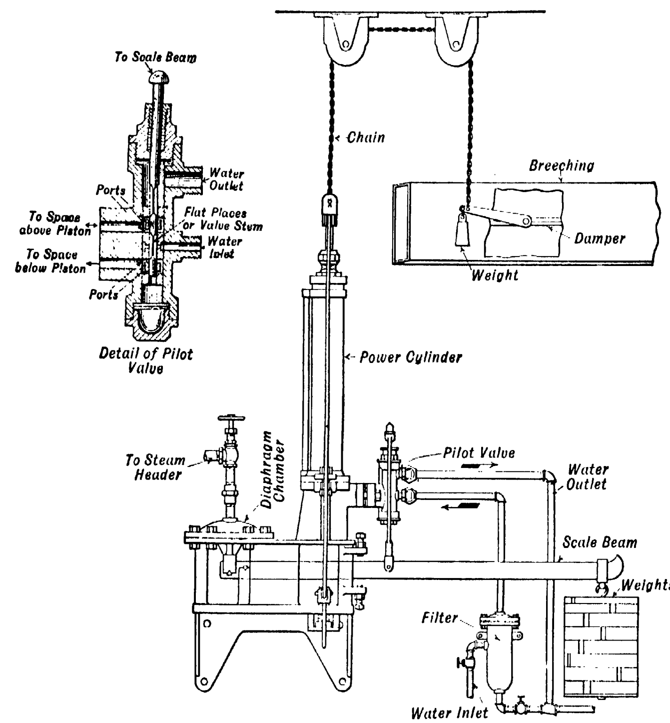


FIG. 186. — R-K Double Acting Hydraulic Damper Regulator.

sure, the counter-weight moves the pilot valve to discharge the operating fluid from the power cylinder and increases the opening of the damper.

The steam-operated regulator has steam pressure acting directly on the power-cylinder piston, which moves against a spring of the proper strength. The piston rod is connected directly to the damper by suitable means.

REFERENCES

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Steam Power Plant Engineering, GEBHARDT.
Chimneys, CHRISTIE.

REVIEW QUESTIONS AND PROBLEMS

1. Deduce, from fundamental considerations, an equation for the height of a chimney.

2. Using Kent's equation, determine the height of a chimney suitable for 3000 boiler horsepower, assuming a diameter of 144 inches.

3. Find the height of a mill chimney having 1.16 inches draft at its base, if the average inside temperature is 400 deg. fahr., and the outside temperature, 60 deg. fahr. Also find the boiler horsepower, which the chimney would be adapted for, if the inside diameter were 9 ft.

4. What is the assumption regarding the rate of combustion per boiler horsepower per hour, in Kent's equation?

5. Describe the construction of (*a*) a brick chimney, (*b*) a steel chimney.

6. Name the two systems of mechanical draft. Which system is commonly used on shipboard?

7. Name the types of draft fans. Which is the most efficient type?

8. Describe the operation of a balanced draft system.

9. Explain how a damper regulator operates.

CHAPTER XV

COAL AND ASH HANDLING EQUIPMENT

297. Foreword. — In small power plants, coal and ashes are moved by hand. Coal is shoveled into a wheel-barrow or small truck at the storage pile, moved to the front of the boiler and dumped on the floor or fired directly into the furnace from the barrow. From the floor it is shoveled into the furnace or hopper of the stoker, according to the method of firing employed. The ashes are raked from beneath the grates to the front of the boiler and shoveled into wheel-barrows or dump cars. The cost of these operations varies, and, in the case of coal, is generally about 50 cents per ton; in the case of ashes, the cost is difficult to determine, because the men performing this work also perform other duties.

In power plants where the quantity of coal handled is large, the expense of handling coal and ashes by the above method would be excessive, and therefore conveyors are used. The conveyors receive the coal directly from cars or outdoor storage, and carry it to overhead bunkers, from which it falls by gravity into the hoppers of the stokers. The cost of handling a ton of coal by means of a conveyor varies from 4 to 20 cents per ton, and the cost of handling ashes varies from 5 to 25 cents per ton.

In general, coal and ashes should not be handled by the same conveyor. When they are handled separately the conveyor system is more flexible and has a longer life. Coal is graphitic and causes small wear of the conveyor parts, while ashes are abrasive and cause excessive wear of the conveyor machinery. Furthermore, the ashes may be hot or wet. When hot they produce distortion and rapid disintegration of the conveyor buckets; when wet they produce corrosion, which results from the formation of sulphuric acid.

298. Classification of Coal Conveyors. — Conveyors used for moving coal may be classified as follows:

1. Scraper, or flight.
2. Bucket.
3. Belt.
4. Apron.
5. Grab bucket.
6. Telpherage systems.

299. Scraper, or Flight Conveyor. — This type of conveyor is used extensively for conveying coal horizontally and for inclinations up to 45

degrees. It is a simple method, but is subject to maximum wear, because the material is pushed and not carried. Its capacity may run as high as 100 tons per hour.

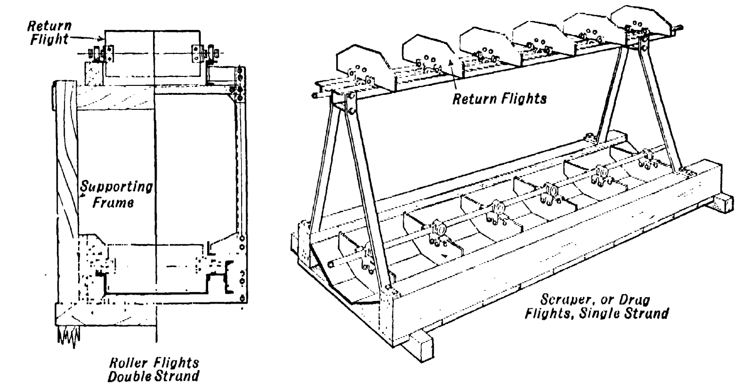

FIG. 187. — Flight Conveyor.

It consists of one or two strands of chain, to which are attached steel scrapers, or flights, which scrape the coal through a trough having the same shape as the flight. The coal is discharged from the trough through gate-controlled openings in the bottom of the trough. There are several

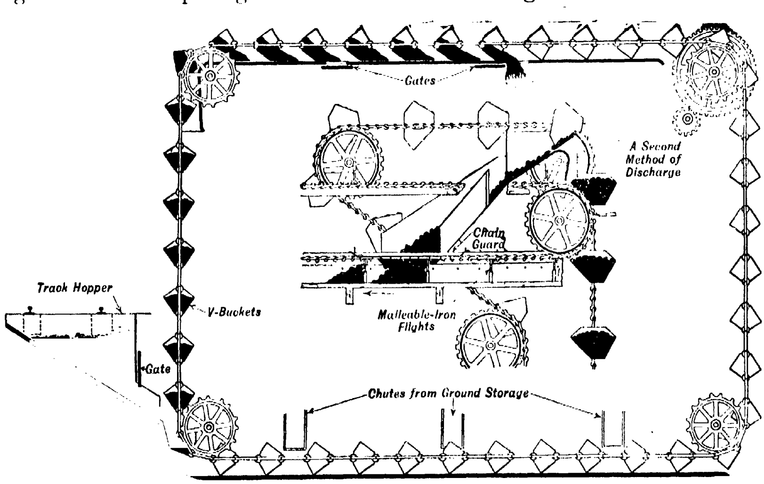

FIG. 188. — Diagram showing Operation of V-bucket Conveyor and Elevator.

types of scraper conveyors, the most common of which are: (1) the plain scraper, Fig. 187, in which the flights are suspended from a chain and drag along the trough; (2) the suspended flight, in which the flights are sup-

ported by a cross bar having wearing shoes at each end, the wearing shoes move along a track and the flight does not touch the trough at any point; (3) the roller flight, in which rollers replace the shoes of the suspended flight conveyor. The last is the most common type.

300. V-Bucket Elevators, and Conveyors. — The V-bucket elevator consists of steel V-buckets rigidly fastened to a continuous steel chain,

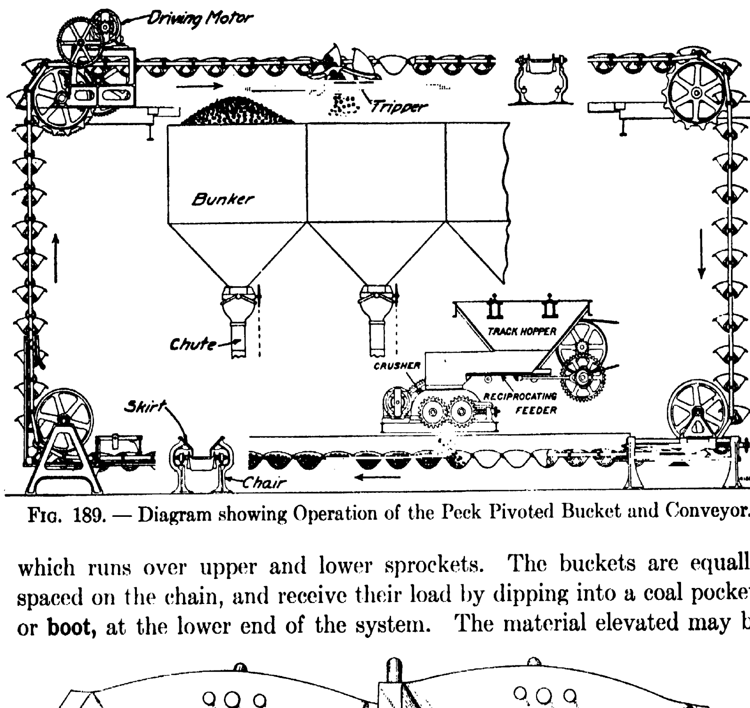

FIG. 189. — Diagram showing Operation of the Peck Pivoted Bucket and Conveyor.

which runs over upper and lower sprockets. The buckets are equally spaced on the chain, and receive their load by dipping into a coal pocket, or **boot,** at the lower end of the system. The material elevated may be

FIG. 190. — Buckets of the Jeffrey Pivoted Bucket Conveyor.

discharged by centrifugal force at the top of the elevator, when the direction of motion is reversed (**centrifugal discharge**), or by the action of a pair of idler sprockets, which draw back the buckets, on the discharge side, at

the top (**positive,** or **perfect, discharge**). In each case the material is positively discharged.

The V-bucket elevator and conveyor, Fig. 188, has the buckets attached to a chain running over sprockets. The buckets lift the coal vertically

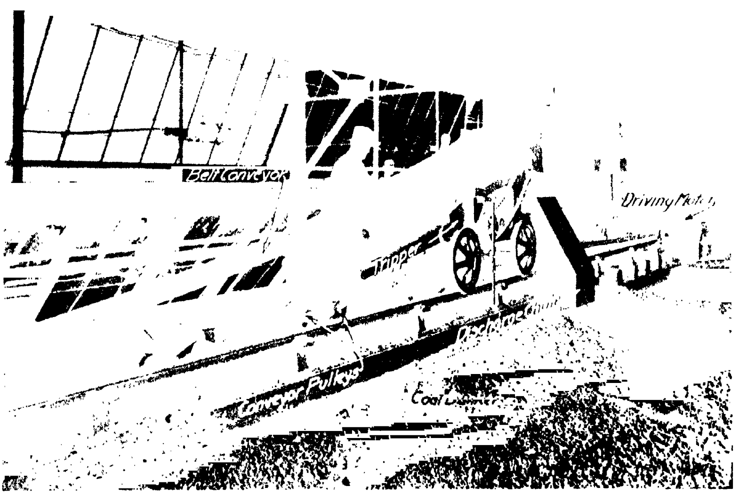

FIG. 191. — Jeffrey Belt Conveyor and Tripper above the Coal Bunkers.

and act as a drag conveyor on horizontal runs, each bucket pushing its half-spilled load ahead of it in a trough of suitable shape. The discharge, on horizontal runs, is through openings controlled by hand-operated gates. The capacity is ordinarily limited to 90 tons per hour.

301. Pivoted Bucket Conveyor. — This type of conveyor, Fig. 189, carries its load in buckets suspended from a pivot shaft, and can convey or elevate, as desired. Its initial cost is high, but the expense of operation is low. It is made in capacities up to 350 tons per hour.

The conveyor consists of a continuous series of malleable iron buckets, Fig. 190, suspended by pivots midway between the joints of two endless chains, which are driven by a motor located at some convenient point, generally at the top of a vertical rise. The buckets maintain their position by gravity and, when traveling horizontally, are supported by a pair of wheels located at each joint of the chains.

The conveyor is loaded by passing below a coal crusher. A **skirt,** supported by castings, overlaps the buckets at the point of loading and

permits continuous loading without spilling of the coal. The coal is discharged into the bunkers by a stationary or movable tripping device, which engages the cams located on the sides of the buckets and tilts the buckets sufficiently to discharge the coal.

302. Belt Conveyor. — The belt conveyor, Fig. 191, gives a flexible means of conveying coal. It is limited, in the inclination at which coal may be elevated, to about 20 degrees. The driving mechanism for the belt conveyor is simple, and may be located at any point along the length of the belt. Belts 1000 feet long are not uncommon. Five hundred tons per hour can be delivered by this type of conveyor.

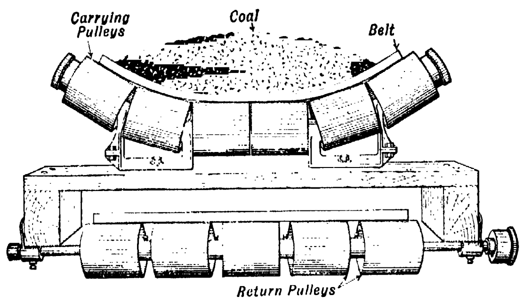

FIG. 192. — Pulley Arrangement of Robins Belt Conveyor.

The conveyor consists of an endless belt traveling over pulleys arranged to give the carrying side of the belt the shape of a trough, as shown in Fig. 192, the return side being supported on flat idler pulleys. The troughing pulleys are spaced from 3 to 6 feet between centers. The coal carried is discharged by movable or fixed trippers, which consist of two pulleys, one above and slightly in advance of the other. The belt runs over the upper and under the lower one, thus causing the coal to fall into a chute which discharges into the storage bunker. A revolving brush is generally placed at the return end of the conveyor to keep the belt clean.

303. Coal Crusher. — When run-of-the-mine coal is to be used in a stoker, it must be reduced in size by passing through a crusher, before it is stored in overhead bins or bunkers. Figure 193 shows a typical crusher, to which coal is generally fed continuously by a reciprocating pusher or apron feeder. The crusher drum shaft is driven from the flywheel shaft by spur gearing, and the crusher drum carries teeth which grasp the coal and draw it between the drum and **crusher plate,** thus reducing the lumps to small size. The crusher plate is grooved for the larger teeth, which are required to grasp and break the large lumps into sizes suitable for the smaller teeth to still further reduce in size.

The crusher is ordinarily located in a pit below the track hopper, from which the coal is delivered by gravity to some form of apron conveyor or reciprocating feeder. The object of the feeder is to provide a regulated

supply to the crusher. If the supply is not regulated, an ordinary two-roll crusher will choke.

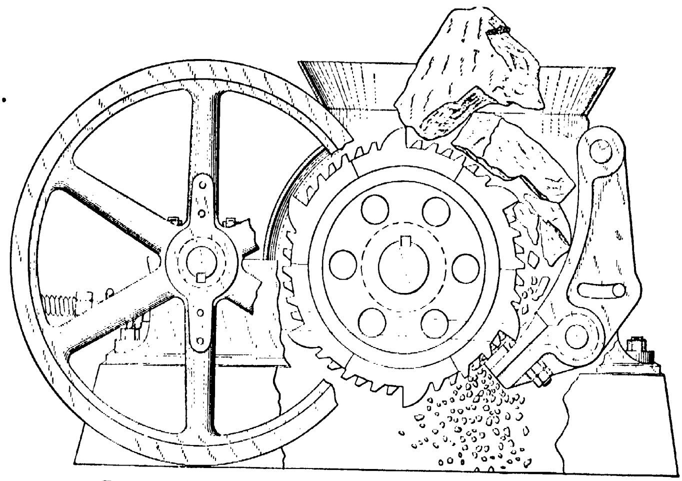

FIG. 193. — Sectional View — Single Roll Coal Crusher.

The **apron feeder,** Fig. 194, consists of overlapping metal slats riveted to two strands of roller chain traveling slowly on tracks, and is used especially when the coal must be lifted in transit or carried some distance to the

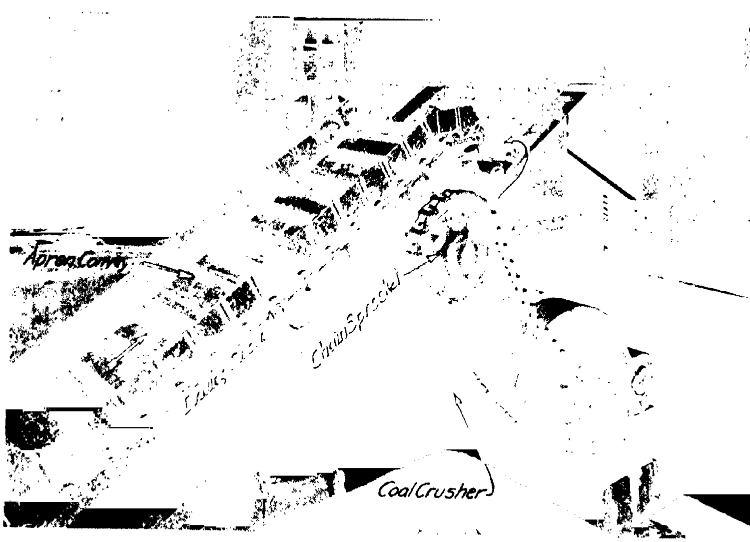

FIG. 194. — Jeffrey Steel Apron Conveyor used to carry Coal from Track Hopper to Crusher.

crusher. The inclination at which this feeder will handle coal is 30 degrees, and only end discharge is possible.

304. Coal Weighing Hoppers and Coal Valves. — Some suitable method of weighing coal is necessary in order that satisfactory records of the quantity of coal burned may be kept. In power plants having overhead bunkers the coal is weighed in stationary or traveling hoppers which have a scale attachment. Coal from the bunker is delivered into the coal-weighing hopper, its weight is recorded, and the coal is then discharged into the hoppers of the stokers through a chute having a flexible joint which permits swinging through several degrees either side of the central position of the chute, at the lower end of which a **coal valve** is used. The hopper mechanism is sometimes arranged to discharge 100 pounds of coal at one time.

Coal valves are made in a variety of shapes and sizes. The simple slide valve and the simplex and duplex rotating valves are used to draw coal from overhead bunkers, and the flap and rotating valves to draw coal from the side of a bin. The simplex valve has a single rotating jaw moved by

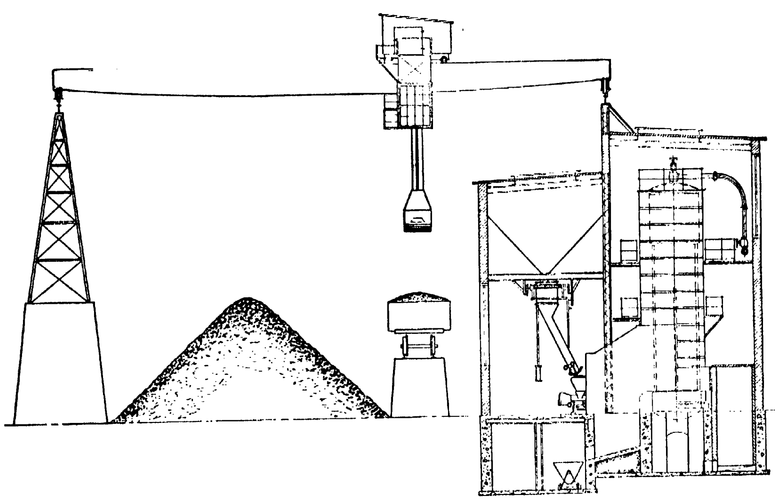

FIG. 195. — Coaling Wickes Boiler House with Overhead Electric Traveling Crane.

a lever, and the duplex valve two rotating jaws, which are moved simultaneously by a common actuating lever. The coal is thus discharged centrally even with a partially opened valve. Both these types cut through the coal without jamming and are easily operated.

The flap valve is the most simple form and consists merely of an iron flap hinged to the bottom of the chute. The flap is lowered to discharge coal and raised to stop the flow.

305. Coal Handling from Ground Storage. — The amount of coal that can be stored in overhead bunkers is limited to a few days' supply, especially in large plants. For instance, the Commonwealth Edison Company in one of its Chicago stations uses 3000 tons of coal in twenty-four hours. In such cases the remaining supply, sufficient for several

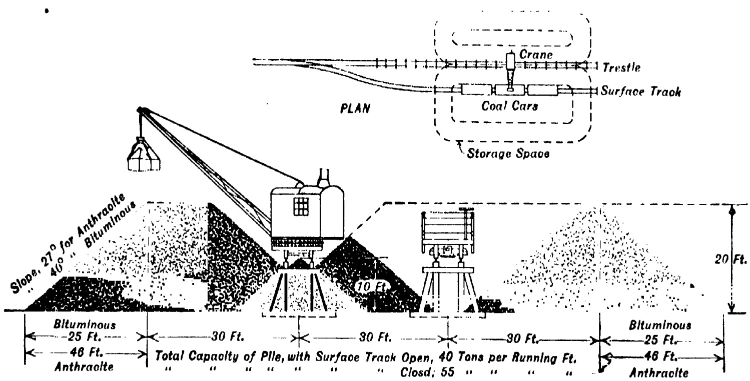

FIG. 196. — Storing Coal using a Locomotive Crane and Parallel Tracks.

months, is stored in large outdoor piles. The above mentioned company stores 300,000 tons in these piles, from which coal is delivered to overhead bunkers as needed. When the storage space is near the water front, coal is delivered in barges, and grab buckets and telpherage systems, such as shown in Fig. 195, are used to remove the coal from the barges and deliver it to the storage piles.

Cars of coal are generally loaded and unloaded by means of a long-radius steam- or electric-driven crane, Fig. 196, having self-filling buckets. Such cranes are capable of moving from 40 to 250 tons of coal per hour, according to the size of the buckets. Coal can be handled at a small cost per ton by this method.

For small plants a portable V-bucket or belt conveyor is used to load trucks or carts from storage. A small gasoline engine is usually the means of driving the conveyor.

306. Overhead Coal Bins, or Bunkers. — The general shape of overhead bunkers is shown in Fig. 197. They are constructed of $\frac{1}{4}$- to $\frac{3}{8}$-inch steel plate and structural shapes, or of reinforced concrete. The **suspension type of bunker** is the cheapest form, because the plates are only subjected to tension and do not require structural members, except for the ends. The shape of this type of bunker is that of a parabola. To permit self-cleaning, the sides are inclined at an angle of 45 degrees and the bottom is made with a hopper.

307. Ash Handling. — The handling of ashes requires a more rugged type of equipment than is used to handle coal. The overlapping bucket

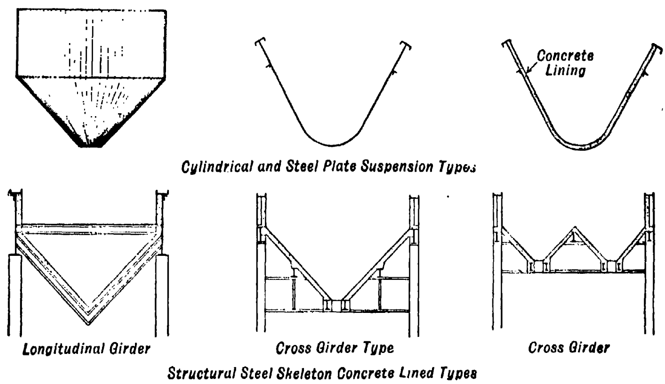

FIG. 197. — Types and Shapes of Coal Bunkers.

conveyor is used extensively, because it carries instead of pushes the material. The other types of conveyors most commonly used are: (1) V-bucket; (2) skip hoist; (3) steam or jet; and (4) drag chain.

308. Skip Hoist. — The skip hoist, Fig. 198, is simple and has few parts at which wear and corrosion can occur. It consists of a bucket running on inclined or vertical tracks, and hoisted by means of a steel cable attached to a winding machine. The bucket consists of a rectangular steel box, open at the top and fitted with guide rollers and hoist bale. In operation, the ashes are delivered into the **skip** by a dump car, and the skip is hoisted by cable and electric motor. At the top the skip is turned upside down and the ashes discharged, after which the skip automatically returns to the bottom for another load.

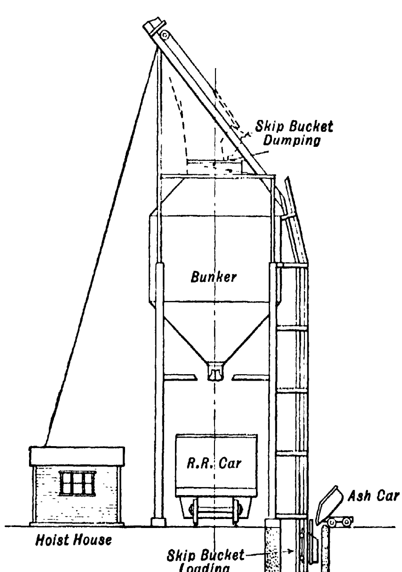

FIG. 198. — Skip Hoist.

309. Steam-jet Ash-conveyor. — The steam-jet conveyor, Fig. 199, consists of lengths of hard extra-heavy cast-iron pipe, held together by means of bolts

which pass through lugs cast on each pipe. Openings into the pipe are made in front of each ashpit, and the ashes are raked into these openings. Expansion joints are used to provide for expansion, and rollers or hangers to support the pipe.

Steam is introduced through one or more nozzles located at an elbow, causing a rapid movement of air through the pipe toward the discharge

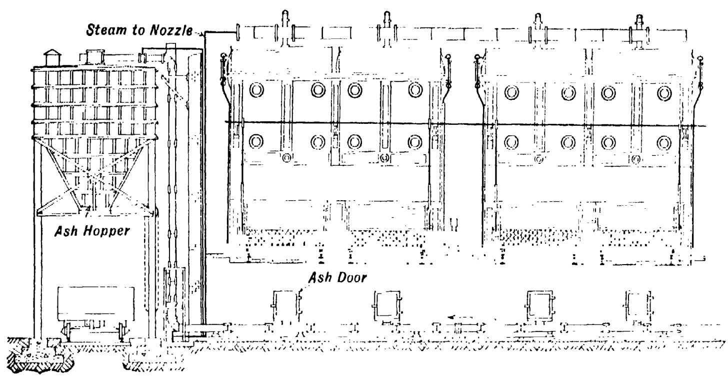

FIG. 199. — Green Steam Jet Ash Conveyor.

end. The moving air carries with it the ashes entering the line at the intakes. The air inlet end of the pipe has a protected hood to prevent clogging of the inlet, and in the discharge pipe provision is made for a water spray to prevent dust. The discharge pipe opens into a **baffle box,** in which the ashes pack and thus eliminate wear, because the discharging ash strikes against ashes. This box also prevents packing of ashes, by breaking the force at which the ashes enter the ash bunker. The elbows where the pipe turns are subject to rapid wear, and are lined with renewable iron inserts which are easily replaced.

310. Drag-chain Conveyor. — The drag-chain conveyor consists of a single-strand grit-proof chain about 7 inches wide, running very slowly in a cast-iron trough. The chain rides on the ashes and does not wear the trough. The cost is low and the space occupied is small.

311. Ash Bunkers. — Ash bunkers resemble coal bunkers in shape and construction. The material coming in contact with the hot, moist ashes must be such that it will not corrode. Steel, wood, concrete, and cast iron are used. When the hopper is in the power house, it generally has a side inlet with a flap valve. A chute delivers the ashes into whatever conveyance is provided to carry them away.

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Power Plant Engineering, FERNALD AND ORROK.
Catalogues of the following Companies: LINK BELT, JEFFREY, LAMSON, STEPHENS ADAMSON.

REVIEW QUESTIONS

1. What is the principal reason for the adoption of mechanical methods of handling coal and ashes?
2. Classify coal-conveyors and describe the construction of a belt conveyor.
3. Describe the coal- and ash-handling equipment in some typical power plant.
4. Name two methods used to handle ashes. Describe one of these methods.
5. Describe the operation of a coal-crusher.

CHAPTER XVI

RECIPROCATING STEAM ENGINE

SIMPLE ENGINES

312. Foreword. — The **reciprocating steam engine** is the most widely known prime mover, and although its field of usefulness has been encroached upon, in recent years, by the steam turbine and gas engine, it is still used extensively. *The steam engine is superior to the turbine for work requiring variable speed, slow rotative speeds and large starting torque, while the turbine has superseded the engine for large central-station units and for auxiliaries requiring high rotative speed.* The high-speed turbine, when used with efficient reduction gearing, has many advantages for low-speed drives, and is rapidly replacing the steam engine in that field. Considering only efficiency in the use of heat, the Diesel type of oil engine is superior to the steam engine; and the turbine is more economical in space requirements. However, the reciprocating steam engine has been enabled to hold its place as a prime mover, by an increase in economy obtained by improved design, that is, by utilizing the uniflow principle in the action of the steam, and balanced poppet valves for high temperatures and pressures. In general, the modern steam engine, in sizes up to 2500 horsepower, is more economical than the turbine.

313. Classification. — Steam engines may be classified in the following ways:

1. By valve gear
   - Slide valve
     - "D"
     - Balanced
     - Multi-ported
     - Piston
   - Corliss valve
     - Drop cut-off
     - Positively operated cut-off
   - Poppet valve
2. By position of the axis of the cylinder
   - Vertical
   - Inclined
   - Horizontal
3. By number of cylinders in which steam expands
   - Single expansion, or simple engine
   - Multiple expansion
     - Compound
     - Triple
     - Quadruple

4. By use { Stationary, Portable, Locomotive, Marine, Hoisting

5. By rotative speed { Low, Medium, High

6. By ratio of stroke to diameter { Long stroke, Short stroke

7. By method of exhausting steam { Condensing, Non-condensing

314. Parts of a Steam Engine. — A simple form of reciprocating steam engine is shown in Fig. 200. Its parts may be separated into the following groups:

1. **Stationary parts:** frame, bed or base, cylinder, cylinder heads, steam-chest cover, bearings, stuffing-box.

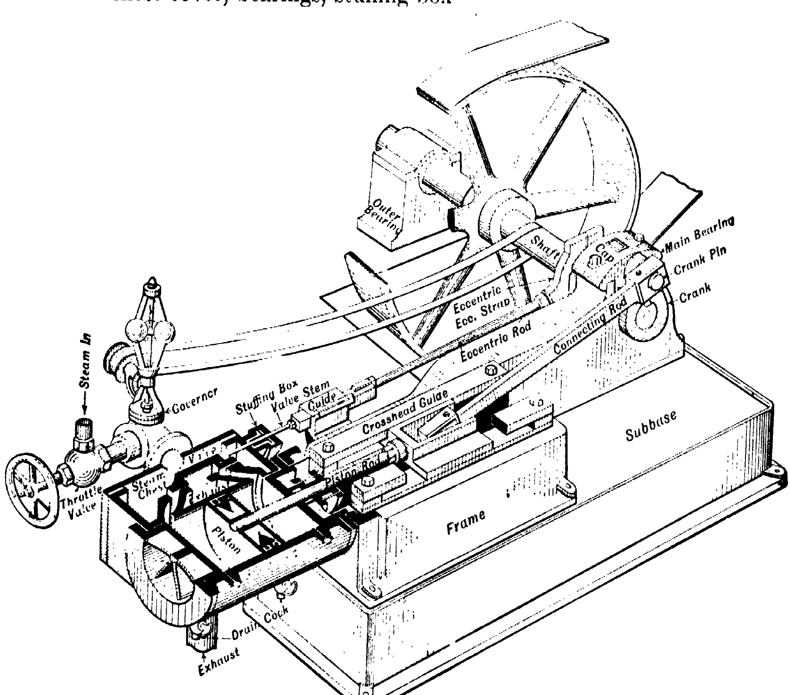

FIG. 200. — "D" Slide Valve, Simple Engine.

2. **Moving parts:** piston, piston rod, crosshead, connecting rod, crank, shaft, and flywheel, all of which are moved by the action of the steam.
3. **Valve gear,** which controls the distribution of steam and consists of eccentric rod, valve rod guide, valve stem or rod, and valve.

315. Function of Engine Parts. — The frame supports and holds the moving parts in proper relative position and gives rigidity to the various moving members. It may rest directly upon the foundation or upon a **cast-iron bed plate** on the foundation. The **bearings** support the shaft and are attached to one end of the engine frame or are supported on a separate foundation. The bearing which is made integral with or rests on the frame is known as the **main bearing,** or **pillow block,** while that resting on a **pedestal** and supporting the outer end of the shaft is called the **outboard,** or **pedestal bearing.** That part of the shaft which turns in the bearing is called a **journal.** The **cylinder** is bolted to the frame at the end opposite the bearings, and forms a chamber in which the piston moves under action of the steam. The ends of the cylinder are closed by **cylinder heads,** and the joint between the cylinder and head is made tight by a **gasket** (a thin piece of asbestos or rubber) or by grinding the surfaces with fine emery to produce a smooth surface, in which case the joint is metal to metal and is called a **ground joint.** The outer surface of the cylinder is covered with **non-conducting material,** to prevent excessive loss of heat by radiation.

The **piston** transmits the pressure of the steam to the crosshead, to which it is attached by the piston rod. At the point where the piston rod passes through the cylinder head, a **stuffing box** filled with packing prevents loss of steam. The **crosshead** slides back and forth, in a straight line, between guides formed in the engine frame. These **guides** make a sliding surface for the crosshead and keep the piston rod from bending. One end of the **connecting rod** is attached to the crosshead and the other to the crank pin, which is fastened to the outer end of the crank and is moved in a circle about the center of the shaft. Hence the connecting rod changes the straight line motion of the crosshead into rotating motion of the crank, which is keyed or pressed on to the shaft to which the flywheel is fastened. The **key** is a rectangular piece of metal which fits into corresponding grooves in two parts of a machine and prevents their relative rotary motion.

The **valve** is located in the **steam chest,** which is ordinarily a part of the cylinder casting and is connected to the cylinder by ports or **passages** made in the casting. It is moved back and forth upon its seat by an **eccentric,** so located on the crank shaft with reference to the crank that it moves the valve to admit or discharge steam to or from the cylinder at the proper time. The **valve rod,** or **valve stem,** connects the valve and

the **valve rod guide,** which is used to prevent bending of the valve rod. The **eccentric rod** connects the valve rod guide and the eccentric.

The **flywheel** has a heavy rim which absorbs energy when the supply of energy is in excess of the demand and gives up energy when the supply is not equal to the demand. It thus prevents rapid fluctuations in speed during a revolution.

The **governor** maintains the speed of the engine nearly constant, by controlling the amount or the pressure of the steam supplied to the engine.

The parts of the engine that move backward and forward in a straight line, such as the piston, piston rod, crosshead, valve and valve rod, are known as **reciprocating parts.** The parts that rotate about an axis, such as the shaft, crank, flywheel and eccentric, are known as **rotating parts.**

The size of a steam engine is ordinarily given by stating, first, the diameter of the cylinder in inches; second, the length of stroke in inches; and third, the number of revolutions as 7 in. by 10 in. — 300 r.p.m.

316. Engine Nomenclature. — The following terms, some of which do not appear to have any logical basis, are applied to steam engines:

Running over is a term applied to the action of an engine when the top of the flywheel revolves away from the cylinder end. **Running under**

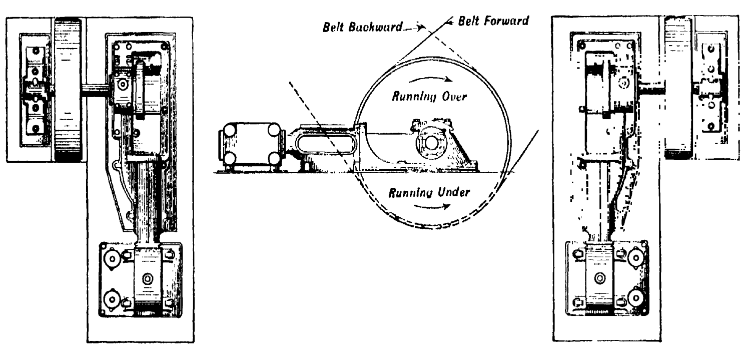

FIG. 201. — Illustration of Several Engine Terms.

is the reverse of running over, the top of the flywheel revolving toward the cylinder end. A **right-hand engine** is one in which the flywheel and valve gear are located on the right-hand side of the steam cylinder, as seen by a person standing at the cylinder end and looking toward the shaft. The flywheel is located to the left of the cylinder on a **left-hand engine.** These terms are illustrated in Fig. 201. The **head end,** or front end, of an engine is the end of the cylinder farthest from the crank. The **crank end,** or back end, is the end of the cylinder that is nearest to the crank and flywheel. The **stroke** is the distance traveled by the engine piston

in passing from the head end to the crank end or vice versa. The **crank throw** is equal to the length of the crank, or one-half the stroke. The **forward** stroke, or out stroke, is made while the piston passes from the head to the crank end. The **return stroke,** or back stroke, is made while the piston travels from the crank to the head end. A **long-stroke** engine is an engine whose stroke is long as compared with the diameter of the cylinder. A **short-stroke** engine has a stroke that is equal to or slightly less than the diameter of the cylinder. **Dead center** is the point at the end of the stroke where the center lines of the connecting rod and crank are in the same straight line. There are two dead centers, one at each end of the cylinder, called the **head-end dead center** and the **crank-end dead center.** **Piston displacement** is the volume through which the piston sweeps in traveling from one dead-center position to the other. *Numerically it equals the net piston area times the length of stroke.* **Clearance** may be either **mechanical** or **volumetric.** The former is the linear distance between the end of the piston and the nearest cylinder head, when the crank pin is on dead center. *Volumetric clearance is the volume of the space included between the piston and the cylinder head, when the crank pin is on dead center, and includes the volume of the steam port for that end. Volumetric clearance is usually expressed in percentage of the piston displacement.* A cylinder having a clearance of 6 per cent would have a clearance volume equal to 6 per cent of the piston displacement.

Example 34. — Compute the volumetric clearance for the head end of a Corliss engine that has a cylinder diameter of 8 inches, a stroke of 24 inches, and a clearance of 5 per cent.

Solution. — Clearance volume, cu. in. = area of piston in sq. in. × stroke in inches × per cent clearance

$$= \tfrac{1}{4}\pi d^2 \times 24 \times 0.05$$
$$= \tfrac{1}{4} \times 3.14 \times 64 \times 24 \times 0.05 = 60.3 \text{ cu. in.}$$

The term **speed,** as applied to engines, refers to **rotative speed,** and is expressed in revolutions per minute. Speeds above 250 r.p.m. are called high speeds; from 150 r.p.m. to 250 r.p.m. medium speeds, and below 150 r.p.m. slow speeds. High-speed engines usually have a short stroke and low-speed engines have a long stroke, compared to the diameter of the cylinder. Piston speeds, expressed in feet per minute, range from 500 to 700. The piston speed is obtained from the formula:

$$S = 2\,LN \qquad (79)$$

in which S = piston speed, feet per minute.
L = length of stroke, feet.
N = revolutions per minute.

Example 35. — The piston speed, in ft. per min. of a 7 in. by 10 in. – 300 Buckeye engine would be

$$S = 2\,LN = 2 \times \frac{10}{12} \times 300 = 500 \text{ ft. per minute}$$

317. Engine Frames. — The form of the engine frame is determined by the type of engine and the purpose for which it is to be used. The frame is made of cast iron, and should have the metal so distributed that the center of the frame is as near the foundation as possible.

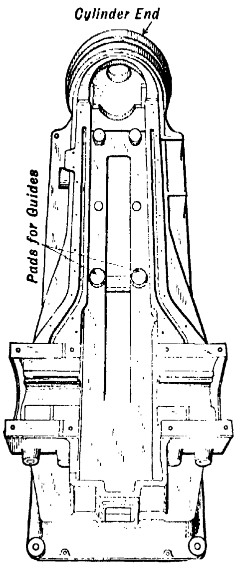

FIG. 202. — Box Frame.

The **box type of frame,** Fig. 202, largely used on high-speed engines, has a rectangular base that rests directly on the bed plate or foundation. The lower crosshead guide is machined as a part of the engine frame, and the upper guide is detachable. The cylinder overhangs one end of the frame and the opposite end carries two bearings, which are separated by a web that stiffens the frame. The recess formed by this web and the side pieces serves to collect any oil that drips from the bearings, and thus keeps it from the floor and foundation. In some types, this chamber is used as an oil reservoir into which the crank dips at each revolution.

The **girder frame,** Fig. 203, is used for Corliss engines doing light work. It is called a girder frame because the part of the frame that connects the main bearing and the cylinder does not rest directly on the foundation, but acts as a girder. The shape of the girder cross section is such that it gives maximum stiffness with light weight. The guides are formed in the girder section.

The **heavy duty,** or **rolling mill, frame,** Fig. 204, has a low center line and heavy construction. The pillow block and guide are made in one casting, thus giving strength and rigidity. The crosshead guides are

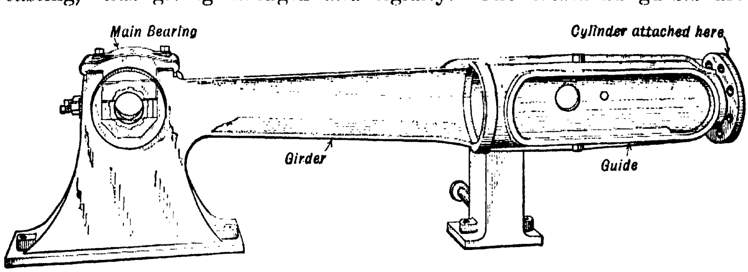

FIG. 203. — Girder Frame — Crank Side.

bored, and the frame is carried under the crank, forming an integral crank case in which oil from the bearings and crosshead guides collects. The oil is then drained from the lowest part of the frame. This frame is of pleasing design and affords continuous contact with the foundation.

A type of frame, Fig. 322, known as the **Tangye frame** is extensively used. The construction is graceful and adapted to heavy duty, since the frame has a stiff back behind the guides.

The frame used for vertical engines generally has the shape of the letter A and is known as an **A-frame.** This frame is usually made with an upper

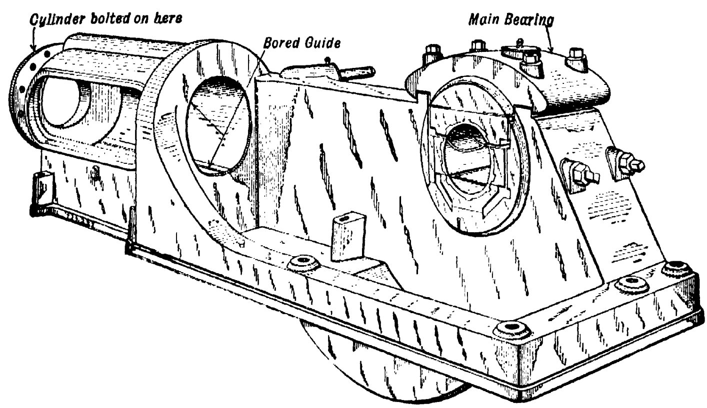

Fig. 204. — Rolling Mill Frame — Crank Side.

and a lower part, the lower part having a rectangular shape and resting directly on the foundation or on a bed plate. A web, cast across the bottom, serves to catch oil. The upper part rests upon the lower, and forms the guides and a support for the cylinder. The vertical frame used on marine engines, Fig. 263, consists of columns, made of cast iron, cast steel, or steel forgings and shaped like an inverted Y. These columns support the cylinder and serve as supports for the crosshead guides. The front and rear columns are often tied together by rods which give sufficient resistance to withstand the severe stresses.

318. Cylinders. — The cylinder and steam chest, Fig. 205, are generally made of cast iron in a single casting, with connecting passages or ports. The body of the cylinder is a shell of uniform thickness, with flanges at each end, to which the cylinder heads are bolted. The diameter of the central portion of the cylinder is called the **bore** and that of the enlarged part at each end, the **counter-bore.** The length of the bore is such that a **piston** ring will overtravel the edge of the bore slightly at each end, and thus prevent the wearing of shoulders in the bore. The counter-bore is

made from $\frac{1}{8}$ to $\frac{1}{4}$ inch larger in diameter than the bore, to permit reboring when the cylinder becomes worn.

The heads are flat pieces, having shoulders which fit into the counterbore. The head-end head is sometimes recessed to permit the piston to

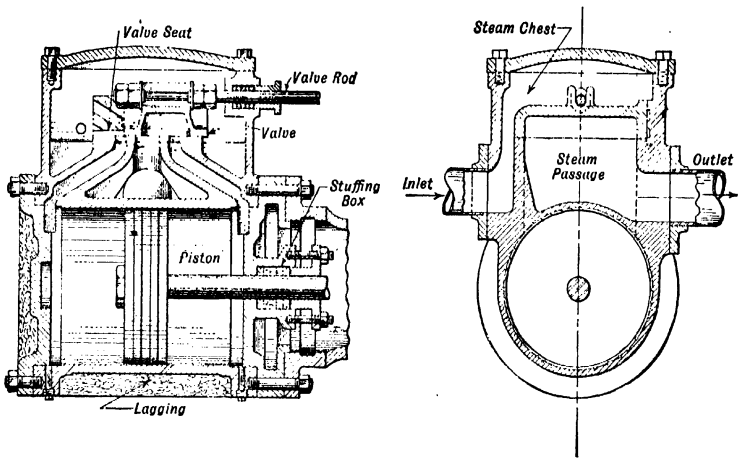

FIG. 205. — Sectional View of Steam Cylinder.

travel closer to the cylinder head, thus reducing the clearance. The crank-end head carries a cylindrical stuffing box through which the piston rod passes. The stuffing box is filled with fibrous or metallic packing which is compressed by a **stuffing box gland** held by studs and nuts.

The part of the casting that forms the steam chest and steam passages is more complicated than the cylinder proper. The **valve seat,** or surface upon which the valve slides, has a rectangular form, and is raised above the surface of the valve chest and machined to a smooth surface. The passages for steam extend from the valve seat to each end of the cylinder; in Fig. 205, these passages are long, and thus increase the clearance. The top of the valve chest is covered with a flat cover plate bolted to the valve chest.

The cylinder is given a covering of planished steel, to improve its appearance, and the space between the covering and the cylinder is filled with asbestos, magnesia, or other insulating material, to prevent loss of heat by radiation. The crank end of the cylinder is machined to fit a corresponding machined part on the frame, and is held in position on the frame by a ring of bolts.

An outside view of a **Corliss engine cylinder** is shown in Fig. 206*a*, and the appearance of the cylinder after being insulated and covered with

planished metal is shown in Fig. 206b. The cylinder is much heavier than the slide-valve cylinder; and it rests directly on the bedplate foundation and is bolted to the frame. The valve seats are circular in section, and extend across the cylinder casting at each end. This permits the use of short steam passages and reduces the clearance. The steam chest is at the top and is separated from the cylinder bore by the cylinder wall, thus serving to **steam-jacket** the upper part of the cylinder. The exhaust-steam passage is at the bottom of the casting and is separated from the cylinder wall by an air space, which prevents heat loss from the cylinder to the exhaust passage. In some types of Corliss engines the cylinder is made with a cylindrical barrel, and the valve openings are placed in the cylinder heads, which are bolted to the cylinder. This makes a simple cylinder casting.

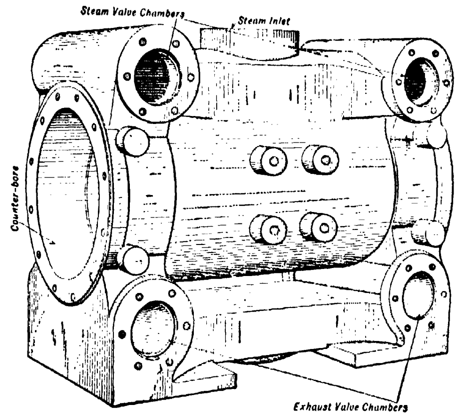

FIG. 206a. — Corliss Cylinder — Outside View.

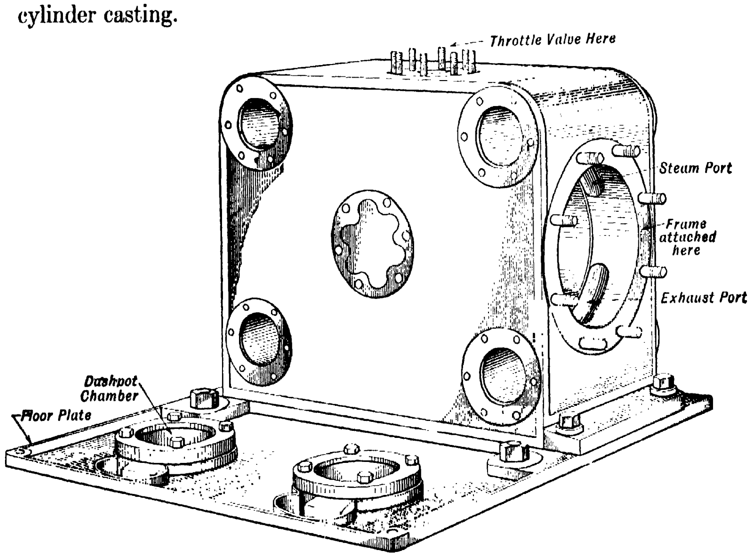

FIG. 206b. — Corliss Cylinder, Floor Plate and Lagging.

The **marine engine cylinder,** Fig. 207, has a liner properly machined and fitted into place, to take the wear of the piston. The cylinder can then be made of a softer grade of cast iron than the liner and, in case of

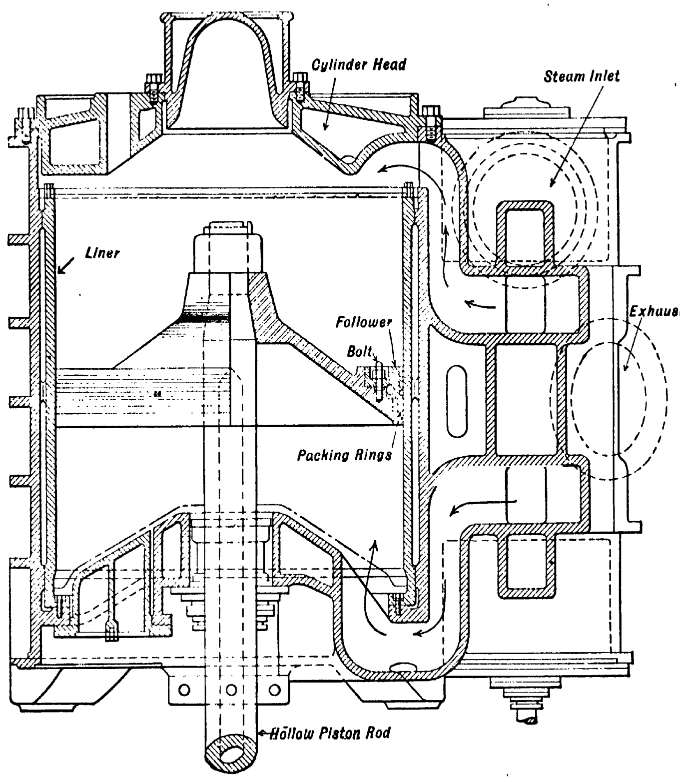

FIG. 207. — Marine Engine Cylinder with Liner.

wear, only the liner need be replaced. The liner is fastened to the lower end of the cylinder by sunk-head bolts. The steam passages are cut through the liner or pass around the ends of the liner, as in the illustration. The heads are double-walled and strengthened by webs to make them strong yet light, and are cone shaped to conform to the shape of the piston, thus reducing clearance.

319. Cylinder Head. — The cylinder head is generally made, like that shown in Fig. 276, of a heavy casting stiffened with ribs. At the point where the steam passages enter the cylinder, the surface of the cylinder head is relieved.

320. Stuffing Box Packing. — The stuffing box used with saturated steam is generally filled with **fibrous packing** impregnated with graphite,

Fig. 208. Leakage of steam past the packing is prevented by tightening the nuts on the gland thus compressing the packing against the rod. A brass bushing is usually placed in the bottom of the stuffing box to prevent

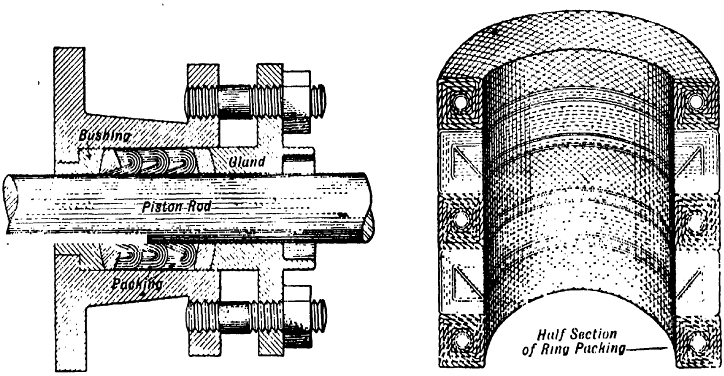

FIG. 208. — Stuffing Box and Several Types of Ring Packing.

the packing from being forced into the cylinder. Fibrous packing often becomes hard and scores the piston rod, because of scanty lubrication. It is not satisfactory for use with superheated steam, because the high temperatures destroy the packing.

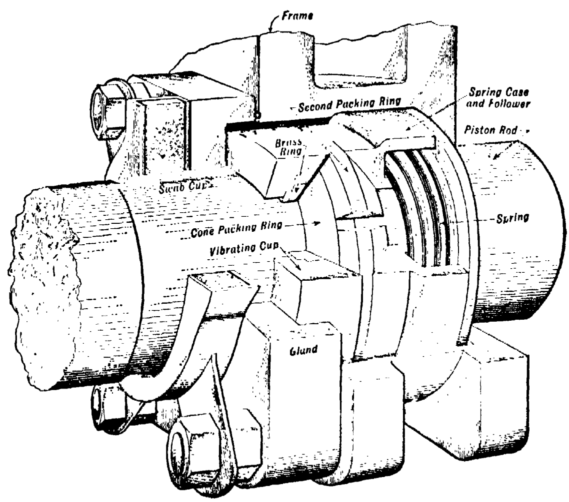

FIG. 209. — Sullivan Metallic Piston Rod Packing.

Metallic packing, Fig. 209, is used with superheated steam. It has long life, has proved satisfactory in service, automatically adjusts itself as wear takes place, and is sufficiently flexible to allow the piston rod to move out of alignment.

A special gland holds the packing in place, and a small copper wire is placed between the gland and the stuffing box to prevent leakage. The

packing consists of a number of beveled rings which are split and fit closely together. The **vibrating cup** surrounds a beveled brass inner ring, made in halves which bear against the piston rod. The **inner packing ring** has a double bevel, one side of which fits the vibrating cup, while the other side fits the **outer packing ring.** The **spring case** fits over the outer packing ring and retains the spring, which holds the various rings in proper

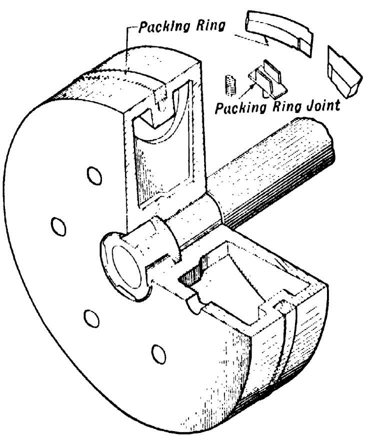

Fig. 210. — Solid Piston with Single Packing Ring.

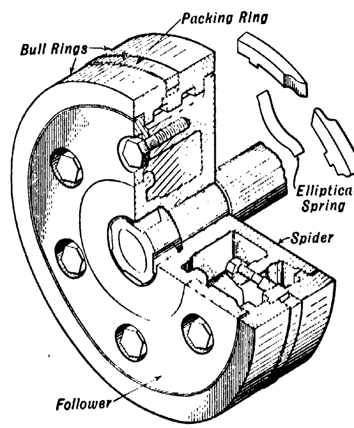

Fig. 211. — Built-up Piston, with Narrow Follower and Elliptical Spring.

position. The **spring** also puts some pressure on the packing, this pressure being increased by the steam pressure acting against the inner end of the spring cage. An **oil-soaked swab cup,** attached to the gland bolts, provides sufficient lubrication.

321. Pistons. — The piston should be of simple construction, of light weight, especially for horizontal cylinders, and strong enough to stand the pressure of the steam without deformation. A piston for a horizontal engine should have a wide face to distribute its weight over a larger surface and thus reduce wear.

The **solid type** of piston, Fig. 210, is quite commonly used. It consists of a hollow cylindrical casting made with flat surfaces of uniform thickness on both sides. A hub, to which the piston rod is attached, is cast at the center of the piston. The hollow part of the casting is divided into several small compartments, by webs which strengthen the end surfaces. Holes are cast in each compartment to permit the removal of the core sand and are plugged after it is removed.

The piston rod is machined to fit into the hub of the piston and is held in place by a countersunk nut screwed on the rod. The nut pulls the hub tight up against a shoulder on the rod, which is often made a straight forced fit into the piston.

Leakage of steam past the piston is prevented by a single packing, or **piston ring,** placed in a groove machined in the circumference of the piston. The ring is made from a piece of cast iron with its outer surface and sides finished. It has a diameter slightly larger than the bore of the cylinder, and is generally made thicker on one side than the other, with a diagonal cut at its thinnest section. The ring is sprung into the groove in the

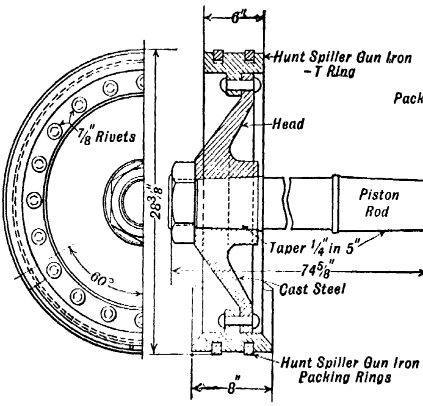

FIG. 212. — Locomotive Piston.

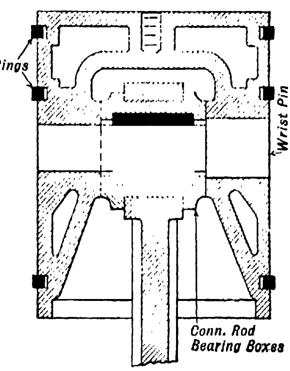

FIG. 213. — Trunk Piston — Sectional View.

piston and, being larger than the bore of the piston, bears lightly against the walls of the cylinder when sprung into place. Leakage of steam through the diagonal cut is sometimes prevented by a tongued clip.

A **built-up piston,** Fig. 211, is much used on Corliss engines. The body of the piston is a flanged spider to which the piston rod is attached. Two **bull-rings,** or **junk-rings,** form the sliding face of the piston, and a single packing ring is placed in the opening between the bull-rings. A **follower plate,** attached to the spider by bolts, holds the bull-ring in position. The position of the bull-rings can be adjusted for wear by set screws fastened to the spider. Elliptical springs are placed at various points between the bull-ring and packing ring. Packing rings are often made in sections, with elliptical springs placed under each section to hold them out against the cylinder bore.

One form of **locomotive piston,** shown in Fig. 212, is made from a simple T-section of cast steel, and has two packing rings. Pistons generally have more than one packing ring, with the rings so placed that the joints do not come in the same line. Leakage past the rings is thus prevented without the use of tongued clips. The piston rod is fastened to the piston by a taper fit, and a nut, placed on the end of the piston rod, holds the piston from coming loose. The nut is generally pinned to prevent its

working loose. Locomotive pistons are often made broader at the bottom, as in the illustration, to increase the wearing surface.

Pistons used on vertical engines are made as light as is consistent with strength. The vertical marine-engine piston, Fig. 207, is made with a narrow face, as the weight of the piston is not carried by the cylinder. The body of the piston is a conical-shaped steel web, and its wearing surface is made up of packing rings of cast iron, held in place by a follower plate bolted to the body of the piston.

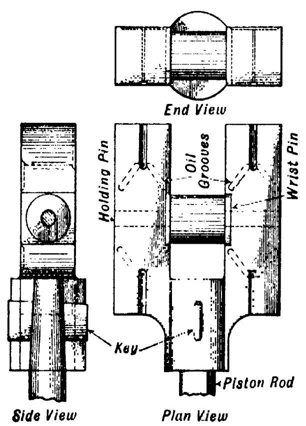

Fig. 214. — Wing Type Crosshead.

On some types of steam engines a trunk piston, Fig. 213, is used. Steam can only act on one side of this piston, which is made long because it performs the function of the crosshead. One end of the piston is open, to admit the connecting rod, and bosses are cast on the inside to support the wrist pin.

322. Piston Rod. — The piston rod is circular in cross section and is made of a good grade of steel, with the crosshead end threaded and the piston end machined to fit a similar machined surface in the piston.

323. Crossheads. — The crosshead is attached to the piston rod by a thread and nut, and to the connecting rod by a crosshead pin, or **wrist pin,** which is carried by the body of the crosshead and which holds one end of the connecting rod. During one-half of each revolution the crosshead **pulls** the connecting rod and during the other half it pushes the connecting rod. This causes a pressure on the guides, which always acts downward when the engine is " running over."

The most common types of crossheads are the " wing," " block," and " slipper." The **" wing,"** or **locomotive type,** Fig. 214, consists of two side blocks, or wings, united by a yoke. The piston rod is screwed into the yoke and provided with a lock nut placed on the piston rod and backed up against the yoke, to prevent the piston rod from working loose in the crosshead, since vibration or removal of the load might loosen the fit between the rod and crosshead and permit the rod to turn out. The pressure of the piston is carried by a steel wrist pin with a flange on each end that fits into recesses in the wings. A small steel pin passes through the wings and the wrist pin, and holds the wrist pin from being lifted vertically. The wings slide between upper and lower guides, which can be raised or lowered by **shims** to provide for wear. The sliding surfaces of the wings are generally made with holes that are filled with babbitt

metal to reduce friction. **Babbitt metal** is an alloy of copper, tin, and zinc or antimony, having anti-friction qualities and a tin content exceeding 50 per cent.

The **" block,"** or **Corliss crosshead,** Fig. 215, has a heavy cast-iron body into which tne piston rod screws. The body is made hollow to admit the end of the connecting rod. The crosshead pin has tapered ends, where it

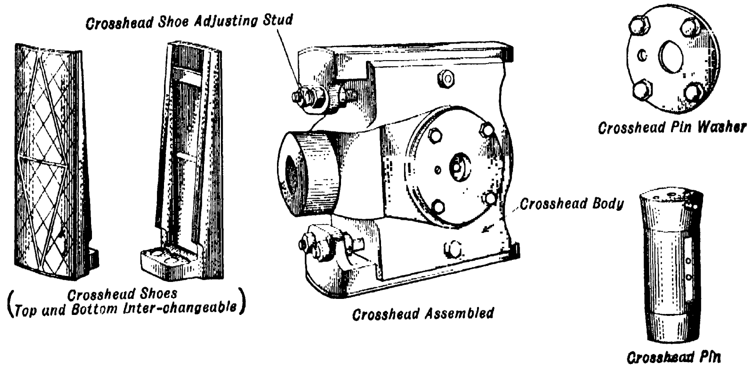

FIG. 215. — Corliss Type Crosshead.

fits into the body, and is held in position by a washer. An **upper** and **a lower shoe,** or slipper, are fastened to the body by side bolts. The surfaces of the shoe and body, where they come together, are made on an incline, and nuts and bolts attached to the body permit adjustment of

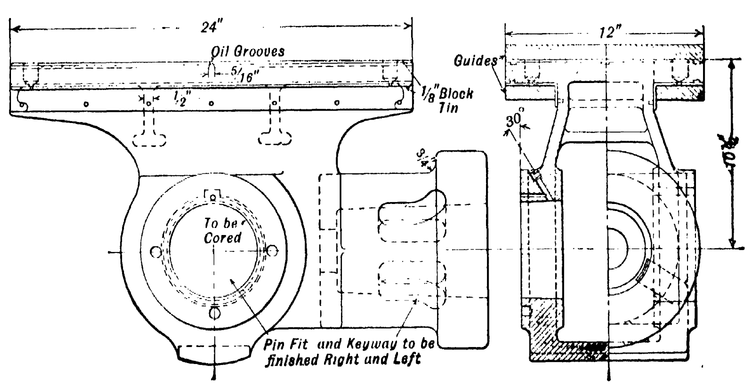

FIG. 216. — Slipper Type Crosshead for Locomotive.

the shoes along the inclined surface. The sliding surfaces of the shoes are babbitted.

The **slipper crosshead,** Fig. 216, is used extensively on marine and locomotive engines. The crosshead body and slipper are made in one

casting, with the piston rod attached to the body above or below the slipper. The sliding surface of the slipper is flat and broad. The guide has a flat planed surface on the engine frame, with adjustable side pieces which fit on top of each side of the slipper and hold the slipper in position. In some types of locomotive crossheads, the guide is a square rod and the crosshead surrounds the guide.

The shape of the sliding surfaces may be flat, circular or triangular. The circular form is the most common and is easily adjusted and machined.

324. Connecting Rods. — The connecting rod is usually made of a good grade of steel, and may have a circular, rectangular or I-section. Corliss and marine engines generally have round rods, and high-speed engines have rectangular rods. Circular rods are largest at the center and taper toward each end. Rectangular rods have the largest section at the crank-pin end.

Common types of connecting-rod ends are the solid, strap and marine end. The **solid-end rod,** Fig. 217, can only be used with a projecting crank pin. It has enlarged and flattened ends, which are slotted to hold

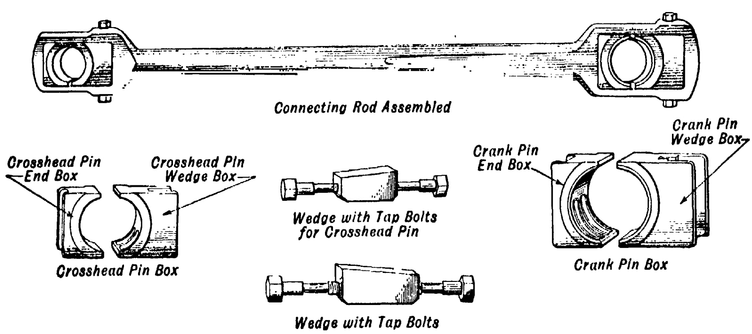

FIG. 217. — Solid End Connecting Rod.

the **boxes,** or **brasses,** in which the crank pin and wrist pin fit. The boxes do not come quite together at the center. This permits adjustment for wear by wedges and bolts placed one inside and one outside, at their respective pins. With the wedges thus located adjustment for wear can be made without changing the length of the connecting rod. The wedges are threaded, and turning the adjusting bolt changes the position of the wedge at right angles to the pin, while side flanges machined on the boxes prevent movement parallel to the axis of the pin.

In the **strap-end rod,** Fig. 218, the brasses are held to the enlarged end of the rod by a strap that passes around them and is held to the rod end by two through bolts. The adjusting wedge is similar to that of the solid-end rod.

The adjusting wedge and through bolts are sometimes omitted and a

metal to reduce friction. **Babbitt metal** is an alloy of copper, tin, and zinc or antimony, having anti-friction qualities and a tin content exceeding 50 per cent.

The **"block," or Corliss crosshead,** Fig. 215, has a heavy cast-iron body into which tne piston rod screws. The body is made hollow to admit the end of the connecting rod. The crosshead pin has tapered ends, where it

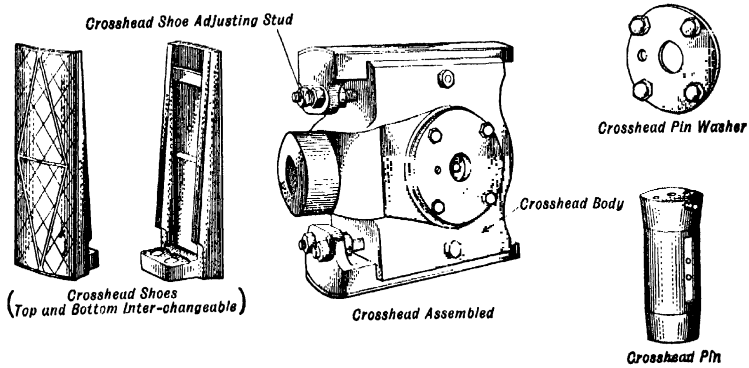

FIG. 215. — Corliss Type Crosshead.

fits into the body, and is held in position by a washer. An **upper** and **a lower shoe,** or slipper, are fastened to the body by side bolts. The surfaces of the shoe and body, where they come together, are made on an incline, and nuts and bolts attached to the body permit adjustment of

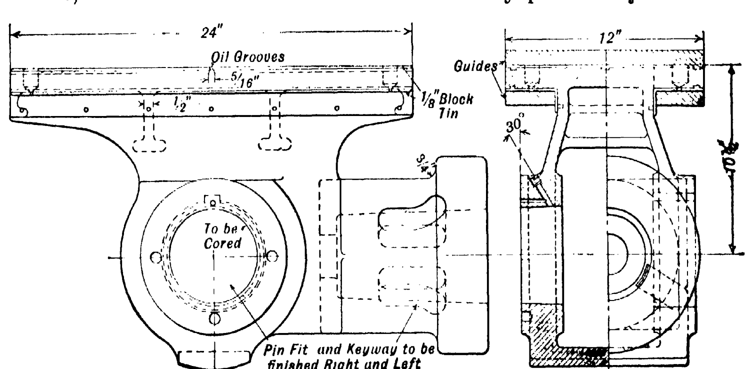

FIG. 216. — Slipper Type Crosshead for Locomotive.

the shoes **along** the inclined surface. The sliding surfaces of the shoes are babbitted.

The **slipper crosshead,** Fig. 216, is used extensively on marine and locomotive engines. The crosshead body and slipper are made in one

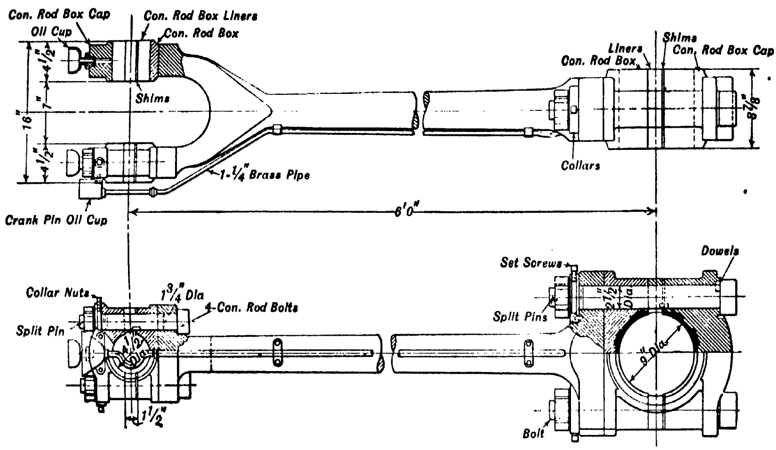

FIG. 219. — Marine Type Connecting Rod.

crank is attached to the end of the shaft, and is often a disk crank; the center crank has the crank pin fastened between two crank disks. Cranks are ordinarily made of cast iron.

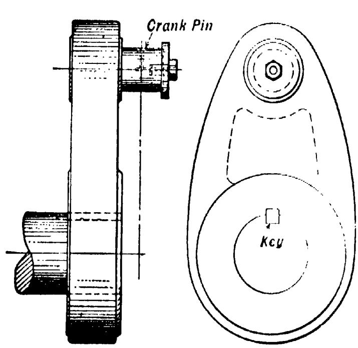

FIG. 220. — Overhung Crank with Crankpin.

The marine engine crank shaft, Fig. 221, has the crank and the crank shaft formed in one piece. It is commonly made in sections, which are bolted together to make a continuous shaft, each section consisting of two crank webs, or disks, a crank pin, and a piece of shaft at each end, which has flanges for coupling. The crank pins are arranged to have the cranks come at 90 or 120 degrees. This gives more even running and distributes the stresses more equally.

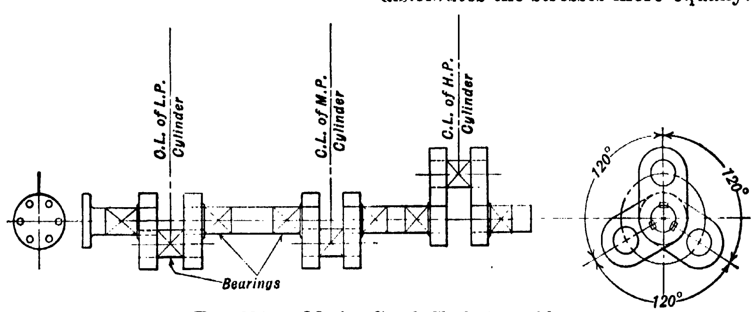

FIG. 221. — Marine Crank Shaft Assembly.

The entire crank shaft is often forged in one piece, and the crank pins and shaft made hollow to decrease weight and increase strength. The locomotive crank is formed as a part of the drive wheel.

326. Bearings. — The simplest form of bearing is the two-part bearing, Fig. 222, generally used for the outboard bearing. It is made of cast iron and consists of an upper and a lower half bolted together and lined with babbitt metal to form the bearing surface. The babbitt lining is ordinarily grooved to facilitate the flow of oil to the bearing surfaces. The lower half is bolted to the foundation. When used as an outboard bearing, it serves to support the weight of the flywheel and keeps the outer end of the shaft in alignment.

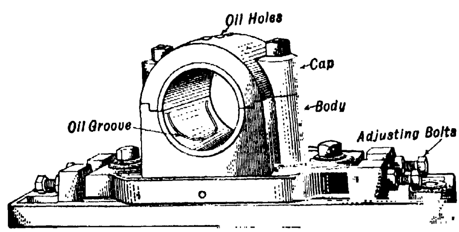

FIG. 222. — Low Type of Two-part Bearing.

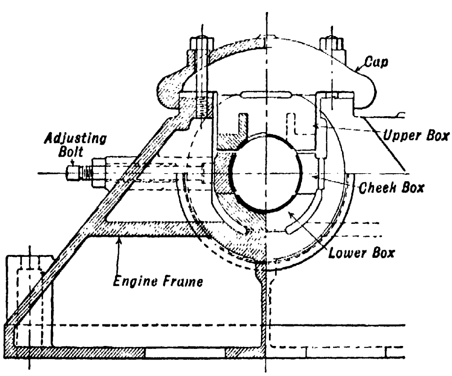

FIG. 223. — Main Bearing, or Pillow Block.

The main bearing, or **pillow block,** Fig. 223, is made heavier than the outboard bearing, to take the thrust from the piston. It consists of a lower part, or **housing,** which may be made separately or as a part of the main frame, and a **bearing cap.** The bearing is formed by three or four cast-iron boxes, which are babbitted and are held in proper position by the lower part of the housing. The boxes are adjusted by inside wedges and bolts which extend through the walls of the housing to the outside. Bolts, which pass through the bearing cap, hold the upper box in position. The lower box is raised or lowered by adding or removing shims from between the box and housing.

A bearing designed to give adjustment at an angle, to compensate for wear, is often used. The opening in this bearing is in a direction normal to that in which the greatest wear occurs.

The bearing, often called a **journal box,** used on the locomotive axle is shown in Fig. 224. The box is made of cast iron and has a brass or bronze bushing which covers only the upper part of the journal. It is held from

rotating by the bearing box. The pressure is always downward on this bearing, and only one bushing is necessary. The lower part of the box forms a cellar, which is filled with oily waste, and also prevents the journal from leaving the box. The locomotive frame fits between projections on the side of the box and prevents the box from moving sidewise.

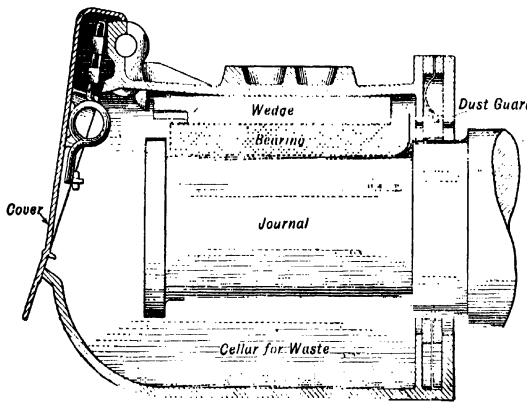

FIG. 224. — Journal Box for Locomotive, — Sectional View.

327. Flywheel.— The flywheel is made of cast iron and consists of a **hub,** a **rim,** and **arms,** which connect the hub and rim. The hub fits over the shaft, to which it is attached by a key. **Set screws** pass through the hub and bear on the key, to prevent axial movement of the flywheel. The hub, rim, and arms may be made in one piece, or they may be made in a number of parts and bolted together. To maintain constant speed, the flywheel should be heavy.

Figure 322, page 413, shows a flywheel made of two pieces, with **through bolts** holding the parts of the hub together, and **stud links,** which fit holes made in the rim, holding the rims together. The length of the links between shoulders, before being put in place, is made shorter than the length between the shoulders in the rim, by a few thousandths of an inch. The link is heated and placed in position; it shrinks when cooling and holds the parts firmly together.

328. Foundation. — In order that an engine may remain level and in satisfactory running condition it is necessary that it rest upon a suitable foundation which may be constructed of concrete, brick, or stone. Concrete is the best material for the part below the ground while hard brick with a granite cap is satisfactory for the part above the ground. When brick or stone are used, they should be laid in a good quality of Portland cement and clean sharp sand. When concrete is used a satisfactory mixture is made of 1 part cement, 2 parts sharp sand and 4 parts gravel or crushed stone.

The shape of foundations varies, wide and shallow foundations being generally preferable to deep and narrow foundations. Straight lines give a more pleasing appearance than curved lines. That part of the founda-

tion which supports the bearings should be tied to the main foundation.

The area over which the foundation should extend is determined by the character of the soil. Foundations built on soils having a low bearing resistance such as clay require a greater area than those built on soils having a high bearing resistance, such as granite rock.

Foundations built on soils of low bearing resistance are generally made in two parts; a sub-foundation and a main foundation. The sub-foundation extends over a greater area than the main foundation, and when made of concrete, its surface is left rough to give a good bond with the main foundation.

Soils which are moist and have low bearing values are generally prepared for the sub-foundation by driving piles, spaced about 3 feet apart, to hardpan or bedrock. The piles are made of red pine, oak, beech, steel, or reinforced concrete, trimmed to an even height and covered with a layer of concrete not less than 24 inches thick, which is often reinforced with steel. The soil directly around the foundation should be well drained.

The weight acting on the soil consists of the weight of the engine plus the weight of the foundation. In general, the weight of the foundation is from 4 to 5 times the weight of the engine.

329. Valve Mechanism. — The various types of valve gears, with their operating mechanism, are described in connection with the engines upon which they are used, Chapters XVII and XVIII.

REFERENCES

Steam Power, HIRSHFELD and ULBRICHT.
Steam Engines, SHEALY.
Steam Power Plant Engineering, GEBHARDT.
Elements of Steam Engineering, SPANGLER, GREENE AND MARSHALL.
Catalogues of NORDBERG ENGINE CO. HARRIS CORLISS ENGINE CO. AND MURRAY IRON WORKS.

REVIEW QUESTIONS AND PROBLEMS

1. Name the parts of a simple steam engine, and state the function of each part.

2. In a manufacturer's catalogue, it is stated that the clearance of the engine is 8 per cent. Explain what is meant. If the size of the engine is 7 in. by 10 in. find the volumetric clearance of the head end.

3. Define: (*a*) right-hand engine, (*b*) stroke, (*c*) dead center, (*d*) high-speed and low-speed engine, (*e*) running over.

4. Calculate the piston speed in feet per minute of an 8 in. by 24 in. Harris Corliss engine, running at 110 r.p.m. Would this speed be greater than a 7 in. by 9 in. Harrisburg engine making 300 r.p.m.?

5. Name three types of engine frames, and compare them with regard to strength.

6. In what way does a Corliss cylinder differ from that of Fig. 205?

7. Name three types of connecting rods, and describe the construction of each.

8. Describe how adjustment for wear is made in a box type of crosshead.

9. Describe the construction of a four-part bearing.

10. State the function of a piston ring, and explain how it performs this function.

CHAPTER XVII

SLIDE VALVE ENGINES, VALVE DIAGRAMS, AND SLIDE VALVE SETTING

330. Foreword. — Slide-valve engines are made in many forms and differ mainly in the type of valve and the method of governing. Most slide-valve engines use a single valve, which may be flat or cylindrical, to control the flow of steam. A few engines use two valves, and some use four valves.

As classified in Art. 313, the most common types of slide-valve engines are: " D " slide valve, balanced, multi-ported and piston.

331. Plain " D " Slide-valve Engine. — The plain " D " slide-valve engine is the simplest of the slide-valve engines, and takes its name from the fact that the section of the valve resembles the letter D. The **valve**

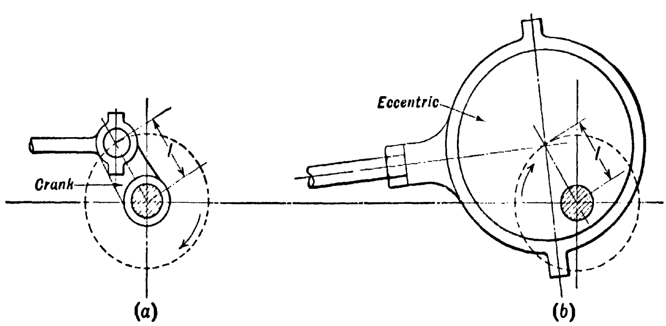

FIG. 225. — Analogy Between Crank and Eccentric.

gear consists of the valve, valve rod, valve-rod guide, eccentric rod, and eccentric, in which the eccentric is direct-connected to the valve.

The eccentric is essentially a short crank, which moves the valve back and forth on its seat. Consider a short crank of length l, Fig. 225a. If the crank pin is enlarged to include the shaft, as in Fig. 225b, an eccentric is formed. The distance between the center of the eccentric and the center of the shaft is the **eccentricity,** or throw of the eccentric, and is equal to l, of Fig. 225a.

The eccentric, Fig. 226, is a circular disk of metal containing a hole which is not in the center of the **sheave** and through which the crank

shaft passes. A set screw generally secures the eccentric to the shaft by which it is revolved. The position of the eccentric on the shaft can be changed by loosening the set screw.

An **eccentric strap,** Fig. 227, made in two parts, surrounds the eccentric. These two parts are held together by through bolts having lock nuts to

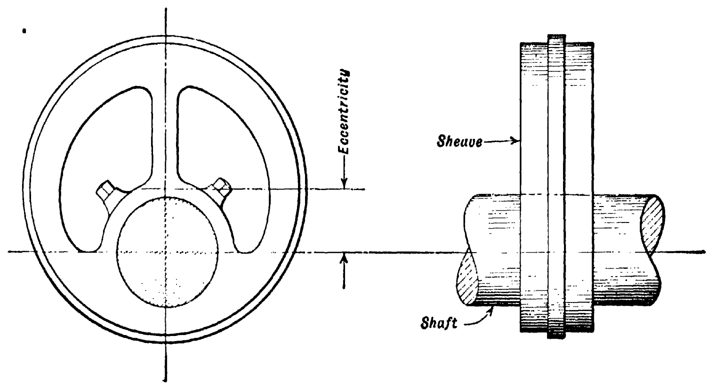

Fig. 226. — Eccentric.

prevent the bolts coming loose. One part of the strap has a hub into which the eccentric rod screws. Side movement of the strap is prevented by a groove in the strap, which fits over a projection on the eccentric.

332. Operation of "D" Slide-valve Engine. — The operation of the plain "D" slide-valve engine may be best understood by first considering what takes place on one side of the piston. Steam, under pressure, enters the steam chest and presses the valve tightly against its seat.

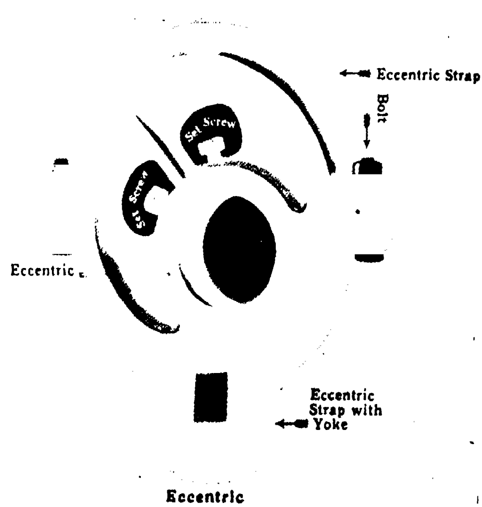

Fig. 227. — Eccentric and Eccentric Strap.

The eccentric is so located on the crank shaft, ahead of the crank in this case, that it will move the valve in the proper direction to admit steam

to the cylinder, just before the piston has reached the end of the stroke. Figure 228 shows the valve in this position, which is called the **point of admission.** The valve and piston are traveling in opposite directions,

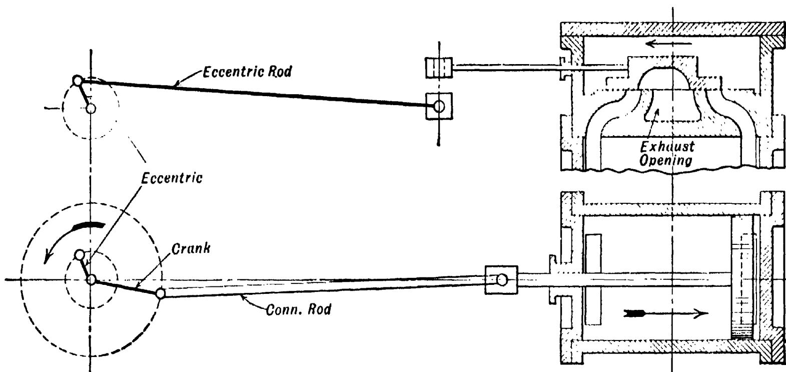

FIG. 228. — Diagram Showing Relative Positions of Crank and Eccentric at **Point of** Head End Admission.

and any further movement of the valve to the left admits steam behind the piston. The pressure of the steam, acting on the piston, pushes it to the left and produces rotation of the crank and crank shaft, to which the eccentric is attached. The valve is thus moved and the steam passage opened wider. Steam continues to enter behind the moving piston, until the eccentric has moved the valve to its extreme right-hand position and returned it to the position shown in Fig. 229. **Cut-off** then occurs, and

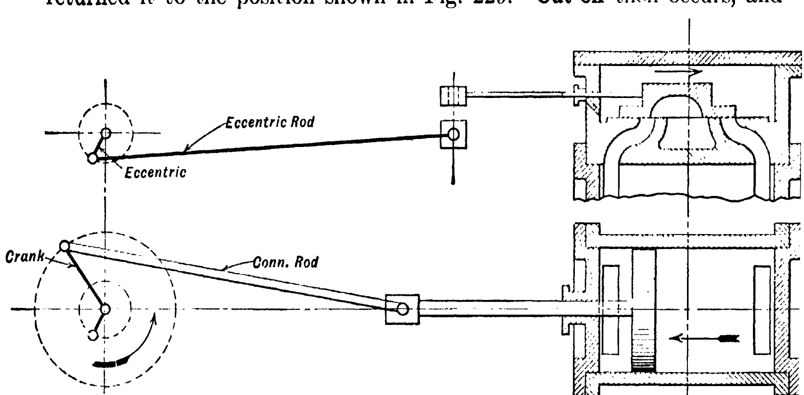

FIG. 229. — Positions of Crank and Eccentric at Point of Head End Cut-off.

steam can no longer enter the cylinder. The valve and piston are moving in opposite directions, and each is traveling opposite to its direction of motion at admission. The steam in the cylinder at cut-off **expands,** that

is, increases in volume, and performs work as the piston moves to the left. *As the volume increases the pressure falls.* When the valve and piston reach the position shown in Fig. 230, **release** occurs, the valve opening the pass-

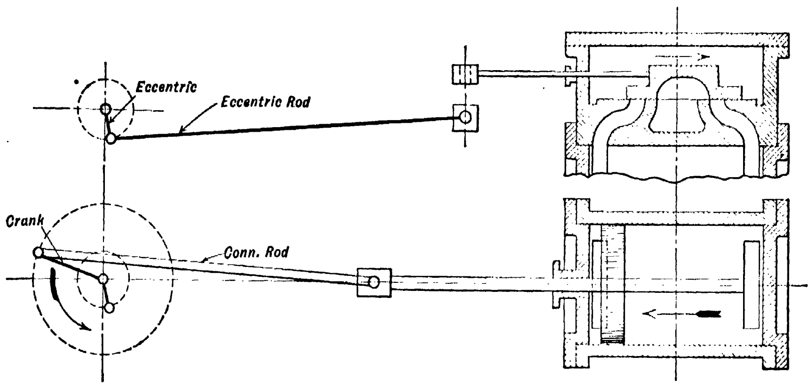

FIG. 230. — Positions of Crank and Eccentric at Point of Head End Release.

age from the cylinder to the exhaust pipe. The exhaust steam passes from the cylinder through the steam passage and the cavity on the under side of the valve to the exhaust opening. At release, the valve and piston are traveling in the same relative direction as at cut-off. Upon further

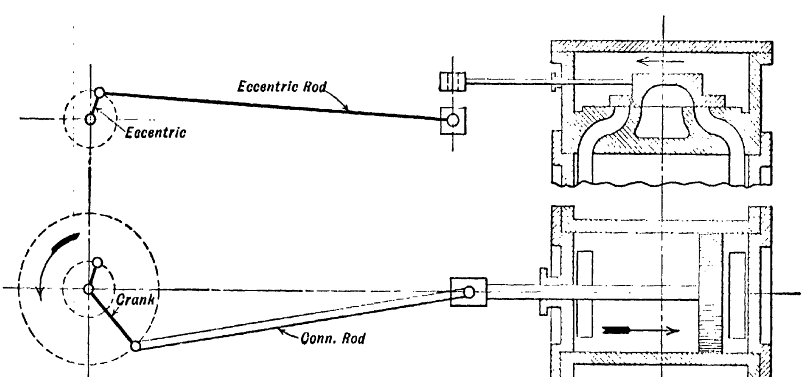

FIG. 231. — Positions of Crank and Eccentric with the Valve at Head End Compression.

movement of the valve, the exhaust opening is increased, and the pressure in the cylinder falls nearly to the pressure in the exhaust pipe. Shortly after release occurs, the direction of the motion of the piston is reversed. It now moves to the right and pushes the steam remaining in the head-end side of the cylinder out through the exhaust passage. When the crank

has nearly reached the head-end dead center, and the direction of valve motion has been reversed by the eccentric, the valve and piston occupy the positions shown in Fig. 231. This is the **point of compression,** and the exhaust passage is closed by the valve. The valve and piston are then traveling in opposite directions.

A **cycle of events** is performed by an engine when it passes through a series of operations and returns to its starting position. *The events in the steam-engine cycle are, therefore, admission, cut-off, release, and compression.* That part of the cycle between the point of admission and cut-off is known as **admission;** the part between the point of cut-off and the point of release is called **expansion;** between the points of release and compression is **exhaust;** and between the points of compression and admission, **compression.** When admission and expansion are occurring in the head-end, exhaust and compression are occurring in the crank end, and *vice versa.*

An engine like the one described, which has steam entering alternately on one side of the piston and then on the other, is called a **double-acting** engine. Most steam engines are double-acting.

If steam acts only on one side of the piston, the engine is known as a **single-acting** engine.

The cycle of events for an engine may be studied by a diagram, Fig. 232, known as an **indicator diagram.** *The vertical height of the diagram*

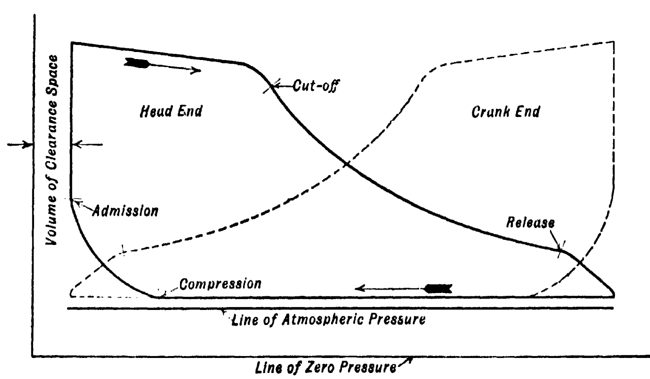

Fig. 232. — Indicator Diagrams.

represents the pressure acting on the piston, to some scale. Horizontal distances represent the volume of the piston displacement at any point in the stroke. The arrows show the direction in which the piston travels. A line representing atmospheric pressure is drawn on the diagram, for reference, to show how the pressure varies as compared with the pressure of the atmosphere.

333. Governor. — The governor used on the "D" slide-valve engine is a **throttling governor.** *It controls the speed by reducing or increasing the steam pressure, depending upon the load to be carried.* The governor operates a throttle valve placed in the main steam pipe, just before the pipe enters the steam chest.

One type of throttle governor, sometimes called the pendulum governor, is shown in Fig. 233. The valve has an upper and a lower disk. Steam at the pressure in the main steam line acts on top of the upper and on the bottom of the lower disk. The same steam pressure acts on the outlet side of both disks. The valve is, therefore, **balanced,** and moves easily. The **valve spindle,** which passes through a **sleeve,** bears against the upper ends of the **governor arms.**

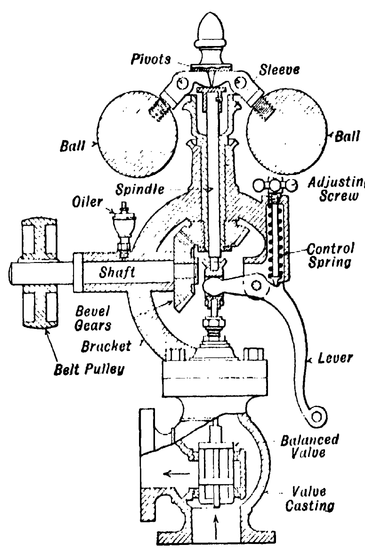

FIG. 233. — Flyball Throttling Governor.

A **bracket** attached to the upper part of the valve body carries a horizontal bearing for the belt pulley shaft, a vertical bearing for the sleeve, and a chamber to which the adjusting lever is attached. The arms supporting the governor balls are pivoted to the upper parts of the sleeve and revolve with it. A **bevel gear** is keyed to the lower part of the sleeve and meshes with a similar gear on the belt pulley shaft. A **belt** connects the **belt pulley** to the engine shaft.

The governor operates as follows: With an increase in the speed of the engine, the speed of rotation of the governor balls is increased. Centrifugal force causes them to move outward and upward against the force of gravity and the force of the **control spring.** The inner ends of the arms of the governor balls thus press down upon the valve spindle and partly close the valve. This reduces the steam pressure acting on the piston, and the speed of the engine is reduced. In like manner, a decrease in speed causes the weights to be lowered and the valve spindle to be raised, thus increasing the speed of the engine.

For any given speed of the engine, the balls will take a position at which their centrifugal force just balances the force of gravity and the tension of the spring. If the engine speed changes, the weights will rise or fall until the steam pressure is adjusted to suit the load to be carried

The speed at which the engine will run can be changed by turning the adjusting screw and thus changing the tension of the control spring. An increase in tension of the spring will increase the speed of the engine and *vice versa.*

A governor, to give satisfactory speed regulation, must have stability; otherwise it would not be able to control the speed, and its movements would be irregular and uncertain.

334. Stability. — The throttle governor already described is **stable;** that is, there is a definite position of the weights for any definite speed. If the speed changes the weights assume a new position corresponding to the new speed. This changes the position of the throttle valve and brings the speed of the engine back to the speed at which the governor is set to run.

335. Sensitiveness and Hunting. — A governor is **sensitive** when a small variation in speed will cause it to move from one extreme of its position to the other extreme. Naturally the more sensitive the governor is, the less stable it will be; but both of these qualities are essential. A governor should not be too sensitive, however, or it will **hunt;** that is, swing first to one extreme and then to the other extreme of its travel, thus making the speed first too high and then too low. Hunting is overcome by using dashpots attached to the governor in such a manner that its motion is damped.

The **dashpot** is a small cylinder having a tight-fitting piston in which there are one or more small holes. The cylinder is filled with heavy oil, which requires time to pass through the holes from one side of the piston to the other and any momentary change of the governor is thus prevented.

336. Terms Applied to Slide Valves. — There are many terms applicable to all types of valve gears, which are more clearly understood when studied with the slide-valve engine.

Valve travel is the distance moved by the valve from one extreme of its motion to the other. When the eccentric is connected to the valve without an intervening **rocker arm,** the *valve travel equals twice the eccentricity.*

Mid-position is the position occupied by the valve when it is halfway between the extreme positions of its motion. *The position of a valve is described by giving its displacement from mid-position.* In most engines, the eccentric is nearly vertical when the valve is in mid-position, or **central.**

Lap is the distance the edges of the valve overlap the corresponding edges of the port when the valve is in mid-position, as shown in Fig. 234. When applied to the inside edges of the valve and port, it is called **inside lap** and when applied to the outside edges, **outside lap.** When steam is admitted to the engine past the outer edge of the valve, the *outside lap* is the **steam lap,** and the *inside lap* is the **exhaust lap.** Steam is often admitted from the inside edge of the valve; the inside lap is then the

steam lap and the outside lap the **exhaust lap.** *Steam lap may be defined as the distance the valve must move from mid-position to allow admission of steam, and exhaust lap as the distance the valve must move from mid-position to allow steam to escape from the cylinder.*

The steam lap is always positive or zero, while the exhaust lap may be positive, negative or zero. When the exhaust lap is negative it is called

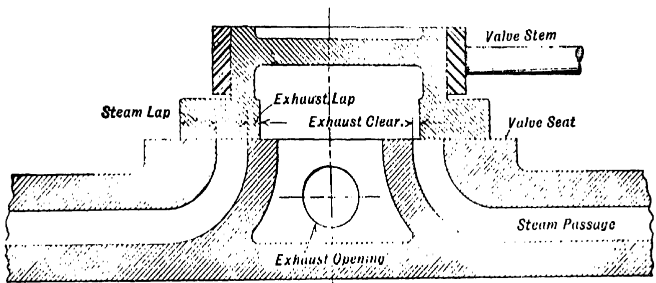

FIG. 234. — Valve Central Showing Laps.

exhaust clearance, and is used to give better distribution of steam. The **angle of lap** is the angle turned through by the eccentric to move the valve a distance equal to the lap.

By referring to Figs. 228 to 231, it will be seen that:

1. When the engine is at admission, the valve has moved from mid-position an amount equal to the steam lap and is moving in a direction to uncover the port.
2. When the engine is at cut-off, the valve is in the same position as at admission but moving in a direction to cover the port.
3. When the engine is at release, the valve is displaced an amount equal to the exhaust lap and moving to uncover the port.
4. When the engine is at compression, the valve is in the same position as at release on that end, but moving to cover the port.

Port opening is the amount the port is open to steam at any instant, and *equals the valve displacement minus the steam lap.* **Maximum port opening** is the amount of port opening at the extreme travel of the valve. This term should not be confused with width of port.

Overtravel is the amount the maximum port opening differs from the width of port, and *equals the eccentricity minus the sum of the lap plus the width of port. It may be positive or negative.*

Lead is the amount the port is open when the crank pin is on dead center. **Steam lead** is given a valve to admit steam to the cylinder just before the piston reaches the end of its stroke. This gives an adequate opening for steam at the beginning of the stroke and assures maximum pressure on

the piston when it is most needed. It also assists the compression in bringing the reciprocating parts to rest without shock, and should be sufficient to give smooth running. It varies in amount from $\frac{1}{32}$ to $\frac{1}{4}$ inch.

The **angle of lead** equals the angle, Fig. 235, through which the eccentric moves to move the valve an amount equal to the lead. It varies from 2 degrees to 8 degrees, depending upon the size and speed of the engine.

337. Angle of Advance. — Referring to Fig. 235, it is seen that *a direct-connected eccentric must be moved ahead of the crank 90 degrees + (the angle of lap + the angle of lead)* to have steam admitted before the piston has reached dead center. The amount by which this angle exceeds 90 degrees is the angle of advance. *It is the angle through which the center line of the eccentric moves to displace the valve from mid-position, an amount equal to the steam lap plus the steam lead.* The Greek letter delta (δ) is used to designate this angle.

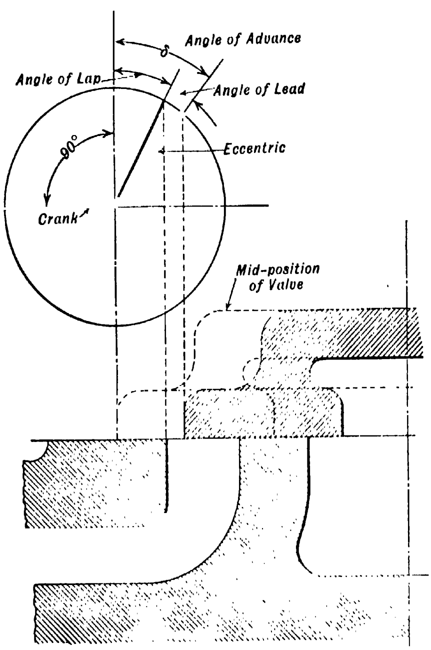

FIG. 235. — Diagram Showing Angle of Lap, and Angle of Lead.

338. The Effect of Lap. — For a valve without lap to admit steam to the cylinder, the eccentric must be 90 degrees ahead of the crank in the direction the engine is to run. Admission would then occur with the crank on one dead center, and cut-off would not occur until the crank had reached the other dead center. Steam would be admitted for the entire length of stroke, and there would be no expansion of the steam. Such an engine would require an excessive amount of steam for each horsepower developed.

Steam lap is given a valve to prevent this waste of steam, and to permit expansion of the steam by bringing cut-off before the end of the stroke.

If there were no exhaust lap, the events of release and compression would occur with the valve in mid-position and with the piston at the end of its stroke, and there would be no compression. To have the exhaust edge of the valve close the port before the crank pin has reached

its dead center position, exhaust lap is given. Compression traps steam in the clearance space and forms a steam cushion, which assists in bringing the reciprocating parts of the engine to rest without shock and makes the engine run more quietly. Zero exhaust lap and negative exhaust lap are given to valves of high-speed engines having a large angle of advance.

339. Relative Position of Crank and Eccentric. — It has been seen that a plain " D " slide valve, connected to the eccentric without an intervening rocker, and having outside admission, has the eccentric ahead of the crank

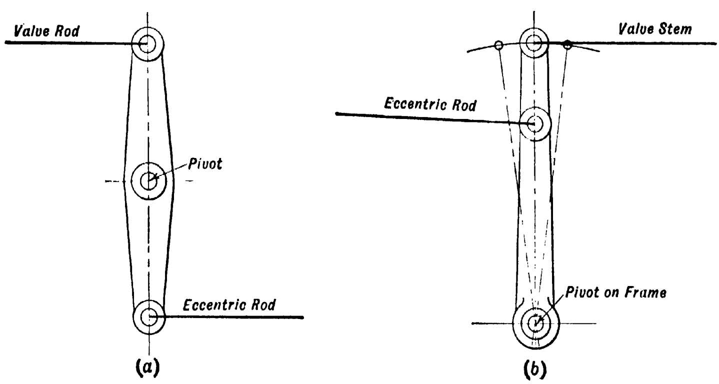

FIG. 236. — Rockers.

by 90 degrees plus the angle of advance. For a direct-connected slide valve having inside admission, the eccentric would follow the crank by 90 degrees minus the angle of advance, to have the valve open the port; otherwise, the valve would be closing the port. In general, the eccentric is located, with respect to the crank, in a position to open the proper steam port when the crank pin moves from dead center in the direction in which the engine is to run.

If a **rocker arm**, Fig. 236*b*, is placed between the eccentric rod and the valve rod, and has a pivot as shown, the rocker arm increases the movement of the valve for a given eccentricity, but does not change the relative position of the eccentric and crank. If the pivot of the rocker arm comes between the eccentric rod and valve rod connection, as in Fig. 236*a*, the eccentric follows the crank for an outside admission valve and leads the crank for an inside admission valve. This type of rocker is called a **reverse lever.**

340. Displacement of Piston and Valve. — *The linear displacement of the piston is described by stating its position from either dead-center position.* This displacement is commonly given in percentage of the length of stroke, and may be found graphically or analytically.

the piston when it is most needed. It also assists the compression in bringing the reciprocating parts to rest without shock, and should be sufficient to give smooth running. It varies in amount from $\frac{1}{32}$ to $\frac{1}{4}$ inch.

The **angle of lead** equals the angle, Fig. 235, through which the eccentric moves to move the valve an amount equal to the lead. It varies from 2 degrees to 8 degrees, depending upon the size and speed of the engine.

337. Angle of Advance. — Referring to Fig. 235, it is seen that *a direct-connected eccentric must be moved ahead of the crank 90 degrees + (the angle of lap + the angle of lead)* to have steam admitted before the piston has reached dead center. The amount by which this angle exceeds 90 degrees is the angle of advance. *It is the angle through which the center line of the eccentric moves to displace the valve from mid-position, an amount equal to the steam lap plus the steam lead.* The Greek letter delta (δ) is used to designate this angle.

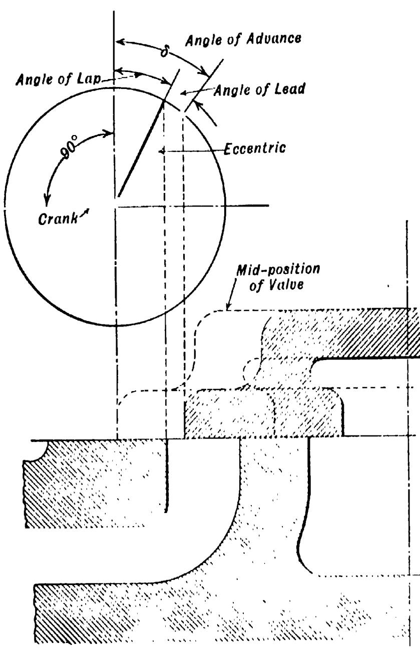

Fig. 235. — Diagram Showing Angle of Lap, and Angle of Lead.

338. The Effect of Lap. — For a valve without lap to admit steam to the cylinder, the eccentric must be 90 degrees ahead of the crank in the direction the engine is to run. Admission would then occur with the crank on one dead center, and cut-off would not occur until the crank had reached the other dead center. Steam would be admitted for the entire length of stroke, and there would be no expansion of the steam. Such an engine would require an excessive amount of steam for each horsepower developed.

Steam lap is given a valve to prevent this waste of steam, and to permit expansion of the steam by bringing cut-off before the end of the stroke.

If there were no exhaust lap, the events of release and compression would occur with the valve in mid-position and with the piston at the end of its stroke, and there would be no compression. To have the exhaust edge of the valve close the port before the crank pin has reached

the piston is represented by the point G. Its position for a connecting rod of infinite length is shown by the point L. The actual position of the piston for the forward stroke is displaced from its position for an infinite connecting rod by the amount LG; for the return stroke this difference is HJ. *For the forward stroke, a finite connecting rod gives a greater movement of the piston for a given position of the crank pin than is shown by the length AL. For the return stroke, a finite connecting rod gives a smaller movement of the piston than is shown by the length BH.* This effect is caused by the **angularity,** or angular position of the connecting rod. The effect is greater for crank pin positions corresponding to the mid-position of the crosshead, and is also greater when the length of connecting rod is short compared with the length of the crank.

Since the eccentric, eccentric rod, valve rod and valve have a motion similar to that of crank, connecting rod, piston rod and piston, the position of the valve may also be found by using a circle having a radius equal to the eccentricity. The diameter of the circle will represent the valve travel. In considering movement of the valve, the angularity of the eccentric rod is neglected, as its length compared with the eccentricity is large, and the valve movement is measured from mid-position, or the vertical axis of the circle.

Since the position of the eccentric and the valve displacement may be represented by using a circle having a radius equal to the eccentricity, and the position of the crank and the piston displacement may also be represented by using a circle with a radius equal to the length of the crank these circles may be superimposed and the movement of the valve, with reference to the piston may be shown by two concentric circles, Fig. 239. *The scale to which the eccentric circle is drawn should be made large, for accuracy of measurement.* The crank position is first drawn at any desired point, and the position of the eccentric located by laying off the angle COe, by which the eccentric is separated from the crank. The above construction must be repeated for each crank position. A more convenient method of studying the valve displacement for all positions of the crank is a graphical construction, termed a **valve diagram.**

The most common forms of valve diagrams are:

Valve ellipse........applicable to any valve.

Zeuner diagram Reuleaux diagram Bilgram diagram	applicable to valves that have harmonic motion.

The **Zeuner** and **Bilgram diagrams** are in common use, and engineers should be familiar with both. *The valve diagram must be interpreted by intelligent reference to the actual mechanism to which it is applied.* A person

should be able, when looking at a diagram, to picture in his mind just what the valve, valve gear, piston and crank are doing at any given time.

341. Valve Ellipse. — This diagram is a closed curve made up of points which represent the displacement of the valve for each corresponding position of the piston. It is drawn by finding the displacement of the valve for each piston position and laying off the displacement, at the corresponding piston position, perpendicular to a center line, which represents the mid-position of the valve. Displacements to the right of mid-position are usually laid off above this center line and those to the left below. A curve drawn through the points thus found will be the valve ellipse. Lines drawn parallel to the center line and separated from it by a distance equal to the steam lap and exhaust lap will cut the ellipse at points corresponding to admission, cut-off, release and compression. A valve ellipse for a Corliss valve engine is shown in Fig. 287, page 366.

342. The Zeuner Diagram. — Referring to Fig. 239, OV is the amount the valve has traveled to the right of mid-position with the crank at C,

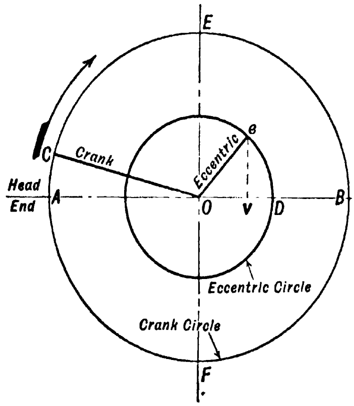

FIG. 239. — Crank Circle and Eccentric Circle Superimposed.

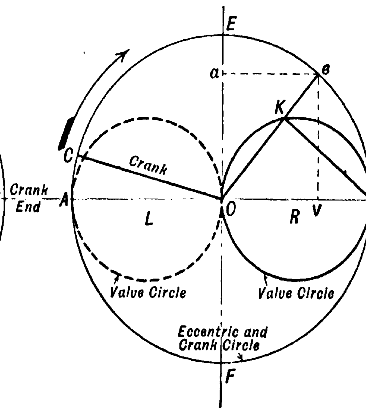

FIG. 240. — Development of Zeuner Diagram.

if the valve has outside admission and the engine runs over. Since the angular displacement of the crank does not depend upon its length, the crank circle may be omitted, and the position of the crank and eccentric represented upon the eccentric circle, as shown in Fig. 240. *The diameter of the eccentric circle then represents the piston travel to one scale and the valve travel to another scale.*

For any crank position, OC, the corresponding position of the eccentric for clockwise rotations of the crank is Oe, and the displacement of the valve is ae. If a perpendicular, KB, is drawn from B to the eccentric position, Oe, a triangle, OKB, is formed which is equal to triangle Oae.

Therefore $OK = ae$, the displacement of the valve. By taking other positions of the crank from A to B it will be found that K will always lie upon the circumference of a circle R, called the **valve circle** and having the eccentricity, OB, as a diameter. The displacement of the valve to the right of mid-position will equal the chord OK cut from the circle R by any eccentric position. For displacements to the left of mid-position there will be a similar valve circle, in the left semi-circle. The latter circle is shown dotted and marked L.

As the relative position of the crank pin and valve are of greater importance than that of the eccentric and valve, the valve circles, R and

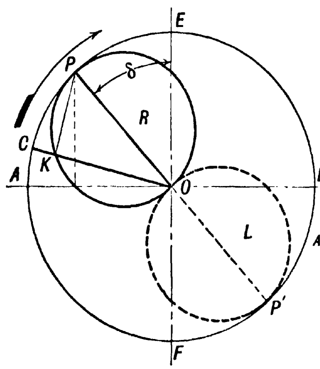

FIG. 241. — Development of the Zeuner Diagram.

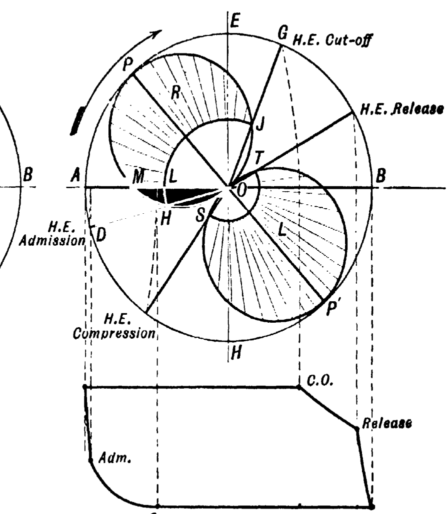

FIG. 242. — Zeuner Diagram Complete for Head End.

L, may be revolved against the direction of rotation until the eccentric, Oe, falls upon the line OC, as in Fig. 241. *The chords now cut from the circle R, by any crank position, will represent the displacements of the valve to the right of its mid-position and the chords cut from L will represent the displacements to the left of mid-position.* **This is the fundamental principle of the Zeuner diagram.** The triangle OKB in Fig. 240 has been turned back through an angle (90 degrees + δ), and the line OB in Fig. 240 now occupies the position OP in Fig. 241. Angle BOE equals 90 degrees and therefore EOP equals the angle of advance.

It has been previously shown that the valve displacement at admission and cut-off is equal to the steam lap. The position of the crank at cut-off and admission can therefore be located on the diagram by drawing an

arc of a circle with O as a center, Fig. 242, and a radius, OL, equal to the steam lap. This arc will cut the valve circle R at the points H and J, and lines OD and OG drawn through these points will represent the position of the crank at admission and cut-off respectively, since the valve displacement OH and OJ is, in each case, equal to the steam lap.

The lead of the valve, which is the amount of port opening with the crank pin on dead center, is represented by the length LM, because the port opening is equal to the displacement of the valve minus the steam lap, and OA is the dead center position of the crank, OM the valve displacement, and OL the steam lap for this crank position. The amount of steam port opening for any crank position equals the valve displacement minus the steam lap, or the distance cut from any crank position by the valve circle and the lap circle, as shown by the radial lines of circle R. The port opening increases from H to P, where it is a maximum, and then decreases from P to J. The port is closed to live steam from J to H. The crank position corresponding to mid-position of the valve is perpendicular to the line OP drawn through the center O, of the eccentric circle.

Since release and compression occur when the valve displacement is equal to the exhaust lap, an arc drawn to cut the circle L at S and T will locate the position of the crank at the point of compression and release. If the valve has exhaust clearance instead of lap, the exhaust lap arc ST would be drawn on the circle R instead of on the circle L, because with exhaust clearance the valve displacement is to the right, instead of to the left, of mid-position for head-end release and compression.

The approximate shape of the indicator diagram, for the above conditions, is as shown in Fig. 242. The points of admission, cut-off, compression, and release are projected to the center line AB, with a radius equal to the length of the connecting rod, drawn to the same scale as the piston stroke AB. The pressure at admission and compression is assumed, and all points projected vertically to the admission and exhaust lines and the indicator diagram, drawn as shown

The valve diagram, Fig. 242, shows only the events that occur for the head end of the cylinder. The same valve diagram may, however, be used to show the events occurring on the crank end of the cylinder, by drawing the crank-end steam lap arc upon the valve circle L, and the crank-end exhaust lap arc upon the valve circle R, as has been done in Fig. 243. This gives a complete diagram, which shows the events for each end of the cylinder. Head-end events are shown with **full lines** and crank-end events with **dotted lines.**

The important points to remember when constructing the Zeuner diagram are:

1. The angle of advance (δ) is laid off from the axis that is perpendicular

to the travel of the piston, in a direction opposite to that in which the crank turns.

2. The chord cut from the valve circle by any crank position shows the displacement of the valve from mid-position.

343. Useful Characteristics of the Zeuner Diagram. — When the *Zeuner* diagram is drawn with the radius of the crank pin circle equal to the eccentricity, it has the following characteristics:

1. A perpendicular dropped from the end of the eccentric position, Fig. 244, always intersects the center line through O, at a distance from

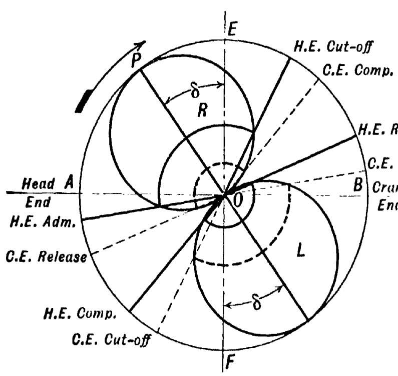

FIG. 243. — Zeuner Diagram for both Head and Crank End.

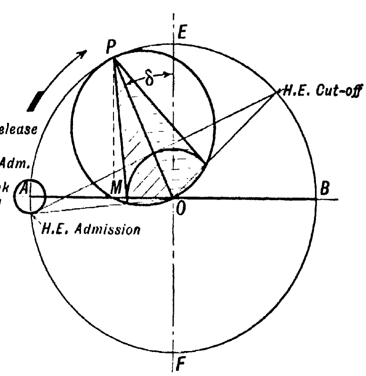

FIG. 244. — Useful Characteristics of Zeuner Diagram.

O equal to the steam lap plus the lead. This is shown by the right-angled triangle OPM, where OM equals the steam lap plus the lead.

2. A perpendicular dropped from P to the crank position for admission and cut-off intersects the valve circle at the same point as the steam-lap arc, Fig. 244. This is also true of the exhaust lap and the crank positions for compression and release.

3. A circle with its center at the dead-center position, OA, of the crank and a radius equal to the lead will be tangent to the crank position at admission. Also, a line connecting the ends of the crank positions at cut-off and admission will be tangent to the lap circle and perpendicular to the line OP.

344. Bilgram Diagram. — Draw a circle with a radius equal to the eccentricity, and a pair of axes as shown in Fig. 245. When the crank is in position OA, the eccentric center line is at Oe. The valve is displaced by an amount which is equal to the steam lap plus the lead and equals ae.

If the crank moves to a new position OC, through any angle α, the eccentric will move to Of, and the displacement of the valve will be $a'f$.

Lay off the angle of advance, δ, from the horizontal axis AB, in a di-

rection opposite to that in which the crank rotates, and call the point on the valve circle, at the end of the eccentric, P. From this point drop a perpendicular Ph upon the crank position OC or OC extended. The length of this perpendicular is the displacement of the valve from mid-position for the crank position OC, because triangle $a'Of$ equals triangle POh and $a'f$ equals Ph. *The fundamental idea of the Bilgram diagram is, therefore,*

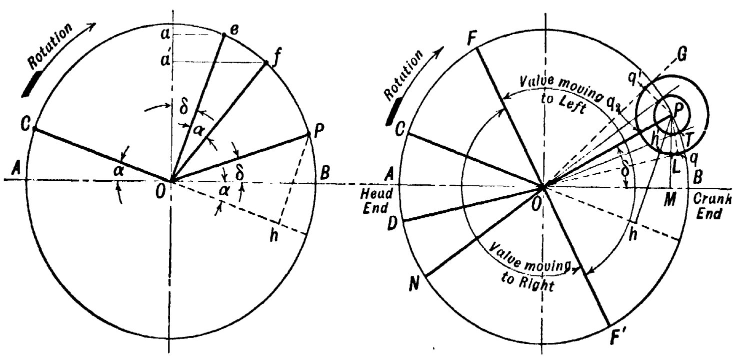

FIG. 245. FIG. 246.

FIGS. 245 and 246. — Fundamental Idea of Bilgram Diagram.

that the length of a perpendicular dropped from P to any crank position, or crank position extended, is the displacement of the valve from mid-position.

The steam-lap circle is represented on the Bilgram diagram by a circle with P as a center and a radius equal to the steam lap, Fig. 246. The exhaust-lap circle is represented by a circle, with P as a center and a radius equal to the exhaust lap. *A negative exhaust lap is shown by a dotted circle.*

The crank position for head-end admission is at OD (the extremity of OD is tangent to the head-end steam lap circle) because the displacement of the valve to the right of mid-position is equal to the steam lap. For the dead-center position, OA, the valve is displaced by an amount PM, which is equal to the steam lap plus the lead, and the port opening is equal to LM, which equals the lead.

For any other position of the crank, OC, the valve is displaced from mid-position by an amount Ph, and the port opening equals hh'. When the crank has moved to OF, perpendicular to OP, the valve has moved to its extreme right-hand position, and from crank position OF to OF' the valve is moving to the left.

When the crank reaches the position OG, tangent to the steam-lap circle, the valve is displaced to the left a distance equal to the steam lap, and the crank is at the crank position for head-end cut-off. At OP the valve is in mid-position.

As the crank moves from OF to OF', the valve is moving to the left, and at T is displaced by an amount equal to the head-end exhaust lap. OT is, therefore, the crank position for head-end release.

At the crank position OF', the valve is at its extreme left-hand position.

Compression for the head end occurs when the crank is at the position ON, because the valve is displaced to the left by the amount of head-end exhaust lap.

A similar diagram can be drawn for the crank end, by repeating this construction with OP extended to cut the valve circle 180 degrees from its position for the head end.

345. Facts Shown by Valve Diagrams. — As the steam lap, exhaust lap and eccentricity are constant for any slide-valve engine, the only change that can be made is in the angle of advance. *An increase in the angle of advance increases the lead and makes all events occur earlier.*

With a constant eccentricity and angle of advance, an increase in the steam lap means a later admission and earlier cut-off. This means a larger valve and increased friction between the valve face and its seat, which is undesirable.

The cut-off for a plain slide valve, having a single eccentric, is limited to half-stroke or later. It is desirable to have as much expansion as possible, and this can only be obtained by increasing the angle of advance, which makes all events occur earlier. Early cut-off is accompanied by small port opening, the effect of which may be counteracted by using multiported valves. The valve diagram shows that the angle turned through by the crank during compression equals the angle turned through by the crank during expansion. Therefore to make cut-off occur earlier, by changing the angle of advance, makes compression occur earlier. An early compression is generally undesirable, as it means a loss of power, except on a high-speed engine, where an early compression is often considered desirable to bring the reciprocating parts to rest without shock.

Example 36. — Given: Steam lap $\frac{1}{4}$ in., exhaust lap $\frac{1}{8}$ in., valve travel 2 in., lead $\frac{1}{8}$ in., no rocker arm, and engine to run over. Find the angle of advance, and the position of the piston at head-end release, compression, admission, and cut-off. Take connecting rod five times the length of the crank.

Solution by Zeuner diagram. — Draw two axes, AB and EF, at right angles as in Fig. 247.

With O as a center, draw the eccentric circle P, with the eccentricity, or one-half the valve travel, as a radius. (*Take twice the size, for accuracy.*)

From O measure off on OA a distance OM, equal to the steam lap plus the lead. (*Double scale because eccen ricity is double scale.*)

Erect a perpendicular at M, cutting the eccentric circle at P; then EOP is the angle of advance.

Upon OP draw the valve circle.

Extend OP to P', and upon OP' draw a second valve circle.

With O as a center, draw a circle having a radius equal to the steam lap, cutting the

valve circle on OP at H and J. Draw crank position through H for admission and through J for cut-off.

With O as a center, draw a second circle with a radius equal to the exhaust lap and cutting the valve circle on OP' at S and T. From O through T, draw the crank position for release, and through S the crank position for compression.

With a radius equal to the length of the connecting rod, swing these points to the line

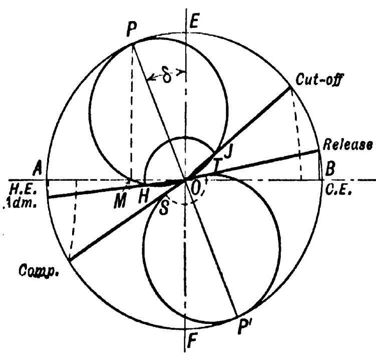

FIG. 247. — Solution of Problem 36 Using Zeuner Diagram.

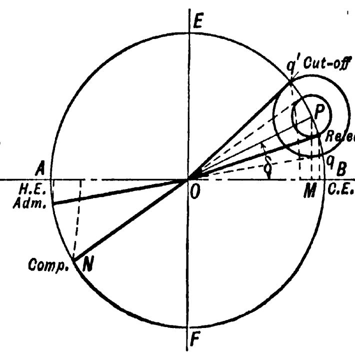

FIG. 248. — Solution of Problem 36 Using Bilgram Diagram.

AB, and measure each from the dead-center position for the end considered. The scale of measurement used is that to which the eccentric circle represents the stroke.

Solution by the Bilgram diagram. — Draw axes AB, EF, and eccentric circle, Fig. 248, as for Zeuner diagram.

To the right of EF lay off a distance PM, equal to the steam lap plus the lead, perpendicular to AB and cutting the eccentric circle. The angle POM is the angle of advance.

With P as a center, draw two concentric circles with radii equal to the steam lap and exhaust respectively.

Draw the crank position for head-end admission tangent to the head-end steam-lap circle at q, and for head-end cut-off at q'.

Draw the crank position for head-end release, tangent to the exhaust-lap circle on the side below OP, and for head-end compression, tangent to the exhaust-lap circle above OP, and extend through O to N.

The remainder of the solution is the same as that based upon the Zeuner diagram.

346. Automatic High-speed Engine. — A typical automatic high-speed engine is shown in Fig. 249. The frame is of the box type, open at the top, and rests upon a cast-iron sub-base. A light, nickel-plated oil shield covers the top opening. The frame is for a center crank and is so made, between the bearings, that it forms an oil reservoir into which the crank pin dips at each revolution.

The valve for the engine shown in the illustration is of the **flat-slide-valve** type and is balanced. The ordinary " D " slide valve has full pressure acting upon its outer surface and pushing its face against the

valve seat. The steam pressure in the ports covered by the valve face reduces the pressure holding the valve to its seat, but the resultant pressure

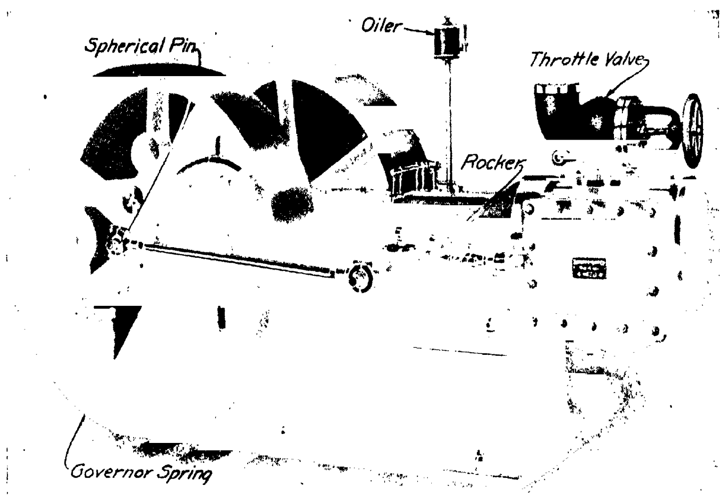

FIG. 249. — High Speed Automatic Engine.

is sufficient to produce a heavy friction load upon the valve gear, that moves the valve.

As there is 60 to 90 per cent less pressure upon the **balanced valve** than upon the ordinary valve, less force is required to move it, and thus

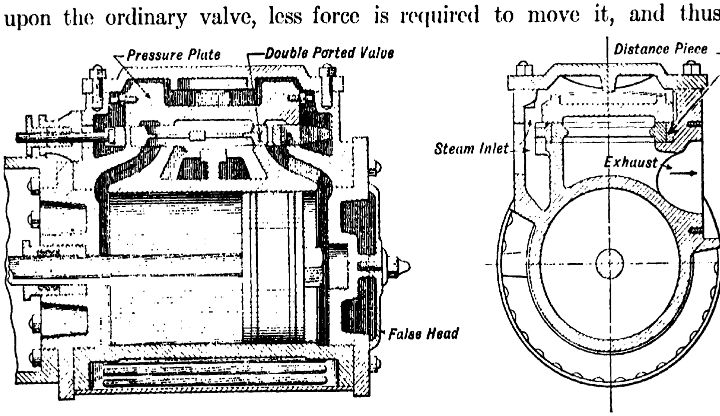

FIG. 250. — Cylinder Showing Sweet Balanced Valve.

the wear is reduced. In one form of balanced valve, used on the Ball engine and shown in Fig. 250, the valve is thin and flat, and slides between a **pressure plate** and the valve seat. The pressure plate rests

upon **distance pieces** at the side of the valve, and keeps the steam pressure from the top of the valve. Springs placed between the steam-chest cover and the pressure plate hold the plate against the distance pieces. Short bolts center the pressure plate and hold it from moving with the valve. This valve is made with ports through it at each end. Steam can pass to the cylinder through the ports and around the edge of the valve, as shown by the arrows. Such a valve is **double ported**; that is, it has two openings for the simultaneous passage of steam. A double-ported valve halves the travel of the valve for a given port opening.

The piston valve shown in Fig. 266, page 347, is commonly used on the high-speed engine. It is a cylindrical slide valve fitted to slide back and forth on a cylindrical seat in the steam chest, and may be single or double ported. As it is generally made, steam is admitted to the cylinder from the inside of the valve and exhausted past the outer edge. In the illustration the valve seat is formed by a thin liner made with slots through which the steam passes. Piston valves are perfectly balanced and are light in proportion to their size. They increase the clearance volume, because of the additional space which surrounds the valve; they make a complicated cylinder casting, and, when wear occurs, they cause increased leakage of steam. To prevent leakage resulting from wear, packing rings are often used, as shown in Fig. 259, page 338, under the description of the locomotive.

The valve gear shown in Fig. 249 differs from the plain "*D*" slide-valve gear previously described, in that it has a rocker arm between the valve rod and the eccentric rod, so arranged that the eccentric, in this case a pin with a spherical end mounted on an arm attached to the governor arm, follows the crank instead of leading the crank.

A **shaft governor** controls the speed, regulating the amount of steam entering the cylinder by changing the point of cut-off. The governor shown in the illustration has a **weighted eccentric arm** supported on the flywheel by a hardened steel **roller bearing** and connected to a **leaf spring** by a **link.** Hardened steel pins and bushings are used to attach the weighted link to the governor arm and spring. *The pull of the spring and the size of the weights determine the speed at which the engine is to run.* To decrease the speed weights are added, and to increase the speed weights are removed.

347. Shaft Governors. — There are many makes of shaft governors, but the underlying principle of all is covered by the types to be described.

The shaft governor is fastened to the flywheel and rotates with it in a plane perpendicular to the shaft.

The position taken by the governor depends on centrifugal and inertia forces, acting in opposition to the pull of one or more springs attached to the wheel on which the governor is mounted.

In all forms of shaft governors, the eccentric is under absolute control of the governor. The eccentric is fastened to the governor in such a manner that any movement of the latter causes a displacement of the eccentric center. Such an eccentric is called a **swinging** or **shifting eccentric,** and in operation changes the point of cut-off, thus proportioning the supply of steam to the load.

The movement of the governor changes the point in the stroke at which cut-off occurs by:

1. Changing the angle of advance.
2. Changing the eccentricity.
3. Changing both the angle of advance and the eccentricity at the same time.

The last method is generally employed with single-valve engines.

Shaft governors may be classified according to the predominating force acting to move the governor as: (1) **centrifugal** governors, or (2) **inertia** governors.

It should be remembered that in all centrifugal governors there are some inertia forces, and that in all inertia governors some centrifugal forces acting on the movable parts.

348. Centrifugal Shaft Governor. — The Armstrong governor, Fig. 251, is a centrifugal governor which swings the eccentric in such a manner that the eccentricity and angle of advance are changed at the same time. The eccentric is not attached directly to the shaft; it is pivoted on the flywheel and has a curved slot through which the shaft passes. An arm attached to the eccentric is con-

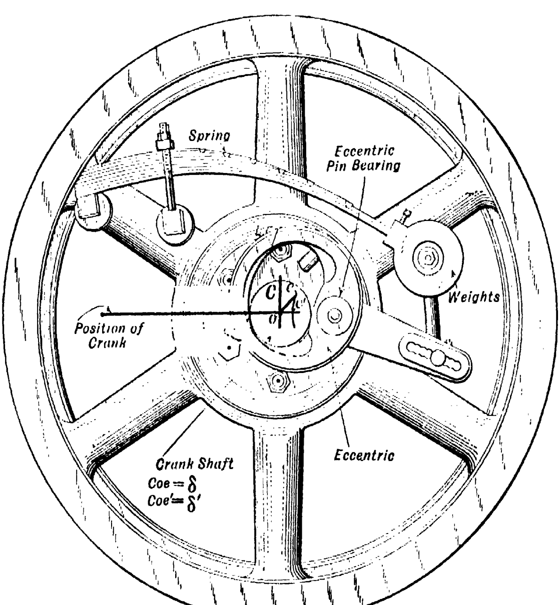

FIG. 251. — Armstrong Centrifugal Governor.

nected to the governor weight, which is fastened to the end of the leaf spring. This spring is firmly fastened to an arm of the flywheel, with bolts. The eccentric is so located that, with the governor weight clear in, cut-off will occur at its latest point.

For the position of the crank pin shown, the eccentric center is at *e* An increase in the speed of the engine causes the governor weight to swing away from the shaft center against the force of the spring. This

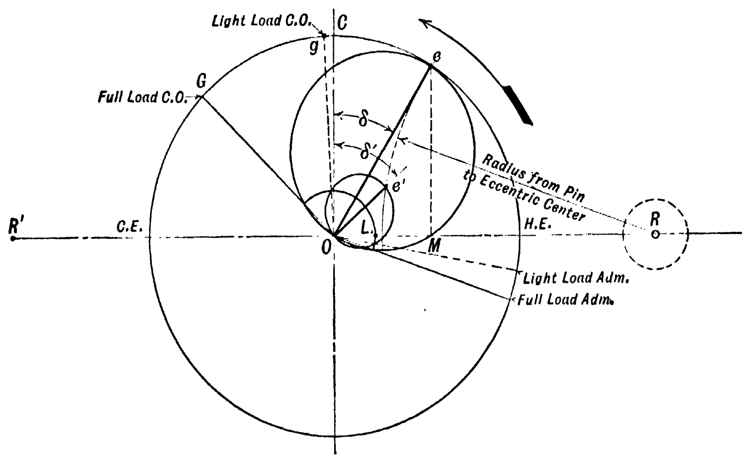

FIG. 252. — Zeuner Diagram Applied to Swinging Eccentric.

action pulls the end of the eccentric arm and revolves the eccentric around its supporting pin. The effect is to increase the angle of advance and decrease the eccentricity.

As the eccentric is revolved about the pin by the governor, because of an increase in load, the eccentric center will travel in an arc *ee'*, with a center at the pivot; at the same time, the angle of advance will increase, as for instance from *Coe* to *Coe'*, and the eccentricity will decrease from *oe* to *oe'*. *When the eccentric center moves in an arc, as shown, the lead of the valve decreases as the speed increases.*

Centrifugal governors are sometimes arranged to move the eccentric center in a straight line across the shaft. The lead then remains constant as the eccentric changes, and the eccentric is called a **shifting eccentric.** An example is the Fitchburg governor.

An illustration of the Zeuner diagram applied to the swinging eccentric is shown in Fig. 252, in which the point of cut-off changes as the eccentricity and angle of advance change. The diagram is lettered to correspond to Fig. 242.

In the Zeuner diagram, the point of eccentric support (R) is shown 180

degrees from its actual position, R'. On the Bilgram diagram this center would be on OC and would be 90 degrees from its actual position, R'. Since the lap does not change while the eccentricity and valve travel change, the steam-lap circle, drawn with O as a center, will cut the valve circles drawn upon the various eccentricities at the points of admission and cut-off.

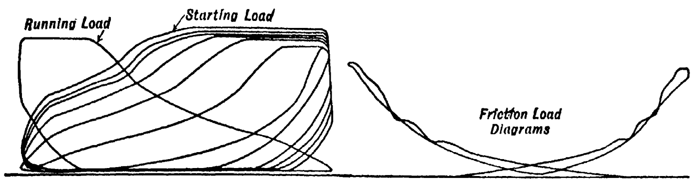

FIG. 253. — Indicator Diagram showing Action of Shaft Governor for a 14 in. by 10 in. Engine.

With the eccentric at e, the cut-off is at G and the lead is LM. When the eccentric is at e', cut-off is at g, earlier than at G, and the lead has decreased nearly to zero.

It should be noted that the position of the pin supporting the eccentric is important. As has been shown, the lead for a swinging eccentric changes as the eccentric center changes from e to e'. An increase in the length of the eccentric arm will keep the lead more nearly constant, which is desirable in order to obtain as much cushioning effect as possible. The lead will always change with this method of moving the eccentric. The amount should, however, be kept as small as is feasible.

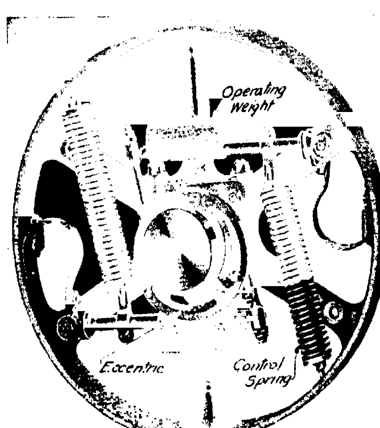

FIG. 254. — Buckeye Centrifugal Governor.

The effect upon the indicator diagram of governing by the swinging eccentric is shown in Fig. 253. This shows that the compression period becomes greater as the cut-off becomes shorter. The compression should be constant to obtain smooth running. On high-speed engines, the changes in compression have small effect because of the large clearance space used.

On an automatic high-speed engine, the valves must be balanced in order that they may be moved easily; otherwise the sensitiveness of the governor will be decreased.

The **Buckeye governor,** Fig. 254, is a centrifugal governor which has coil springs instead of leaf springs. The governor weights are so arranged that they turn the eccentric around the shaft and change the angle of advance, but not the eccentricity. The valve travel thus remains constant. This governor is used with **riding cut-off valves,** and its effect upon the indicator diagram, as the cut-off changes, is shown in Fig. 255.

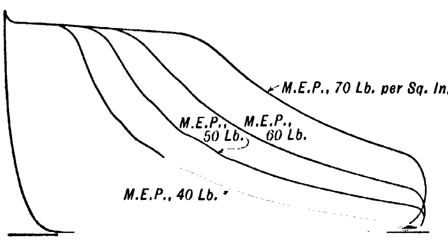

FIG. 255. — Diagram showing Action of Buckeye Governor.

349. Inertia Shaft Governor. — The best example of an inertia governor is the **Rites inertia governor,** Fig. 256. A pin with a spherical end is attached to a heavy arm, carrying a weight at each end, and pivoted to a pin on the flywheel. The pin replaces the swinging eccentric of the Armstrong governor and substitutes for the large sliding surfaces of the eccentric a connection having a small amount of friction and a wider variation of control over the valve motion. The location of the pin is such that the valve motion is the same as for the swinging eccentric, Fig. 251. A coil spring, fastened to the swinging arm and the rim of the flywheel, is the controlling force.

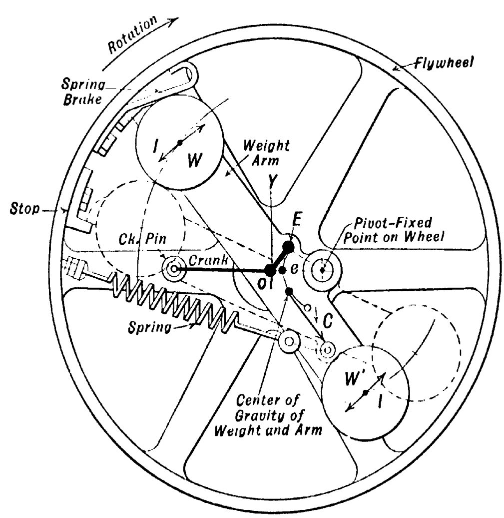

FIG. 256. — Rites Inertia Governor.

The inertia governor is powerful, and acts quickly with change of speed. As the speed increases, the center of gravity of the arm moves outward around the supporting pin under the action of the centrifugal force C, and against the action of the spring. This movement shifts the center of the eccentric from E toward e, and changes the cut-off by reducing the eccentricity and increasing the angle of advance.

Superposed upon this action is the inertia effect of the arm; that is, the tendency of the weights to keep moving at a constant speed when the speed of the flywheel changes. If the load on the engine is reduced, the speed of the flywheel will increase, but the governor arm will continue to rotate at the same speed, because of its inertia. The governor arm will thus lag behind the flywheel. This action moves the eccentric pin nearer to the center of the engine shaft, which decreases the eccentricity and makes the cut-off occur earlier, thus bringing the speed back to normal. The position thus assumed will later be maintained by the centrifugal force, if the new speed is maintained. The particular advantage of the inertia effect is in the rapidity of its action. The Rites inertia governor is simple, but, like the Armstrong governor, does not give a constant lead. It is also unbalanced, as it is pivoted away from the shaft center. For low speeds, this lack of balance affects the point of cut-off and becomes noticeable as a jerk in the action of the governor, which may cause the arm to swing through its whole range every second or third revolution. A drag or brake spring is attached to the flywheel, in such a manner that it bears against one of the weights with sufficient force to prevent this sudden swing, but not to prevent the satisfactory operation of the governor. To prevent the arm from swinging too far, a stop is placed on the flywheel rim.

To secure a balanced inertia governor, two parallel arms, arranged on opposite sides of the shaft center, are sometimes used. This permits running at slow speeds.

350. Locomotive. — The locomotive engine is of the plain slide-valve type. A complete engine, with cylinders and valve gear, is located on each side of the locomotive. The cranks are arranged 90 degrees apart, which is necessary to permit starting when one side is on dead center.

The frame, Fig. 257, differs from the usual type of engine frame in that it is not a solid casting, but is formed by wrought iron or steel **side rails,** which are held together by **cross ties,** or girders. There is a rail on each side to which the cylinders are bolted and which runs the entire length of the locomotive. The cylinder castings are bolted together at the front of the frame to form a **saddle,** or curved portion, upon which the smoke box rests. The smoke box is bolted to the saddle and forms an air-tight connection. The crosshead guides may have one, two, or four bars and are supported at the front by **guide blocks** fastened to the rear head of the cylinder and at the rear by a **guide yoke** which extends across the frame. The guide yoke supports the guides on both sides and is stiffened by a **brace** connected to the guide and riveted to the boiler shell.

Pedestal legs are formed in each side rail to hold the bearing housings

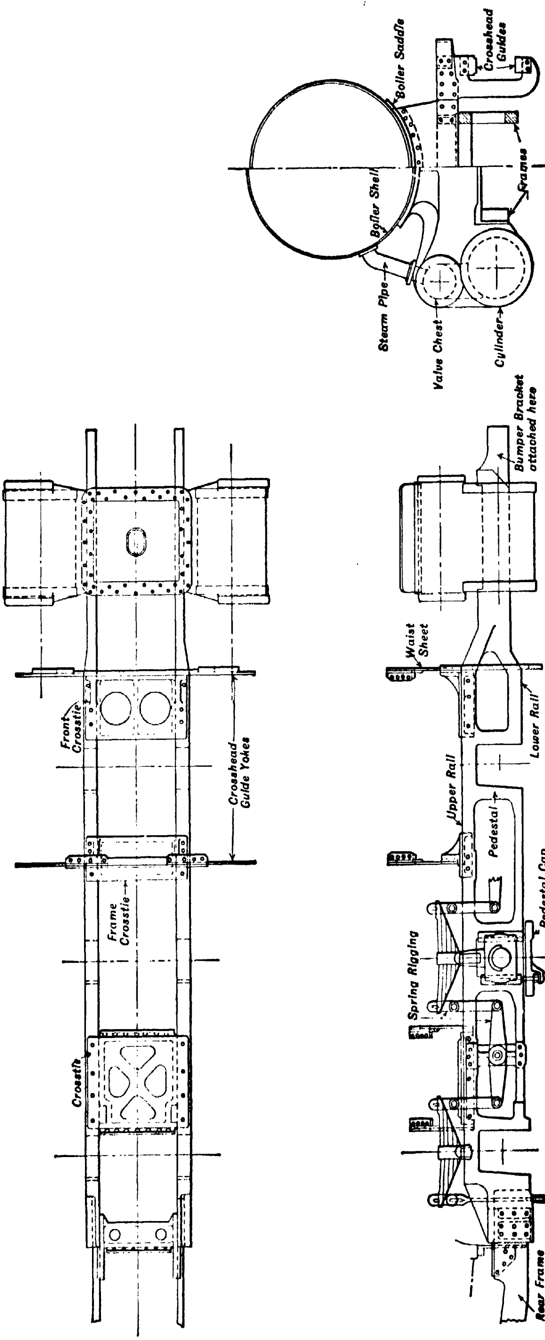

FIG. 257. — Locomotive Frame showing Spring Rigging.

in position. One leg is made at an angle, and a **wedge** is placed between the housing and leg to permit adjustment of the bearing.

The construction of the cylinder casting is shown in Fig. 258. One cylinder and one-half of the saddle are generally made in one piece, but

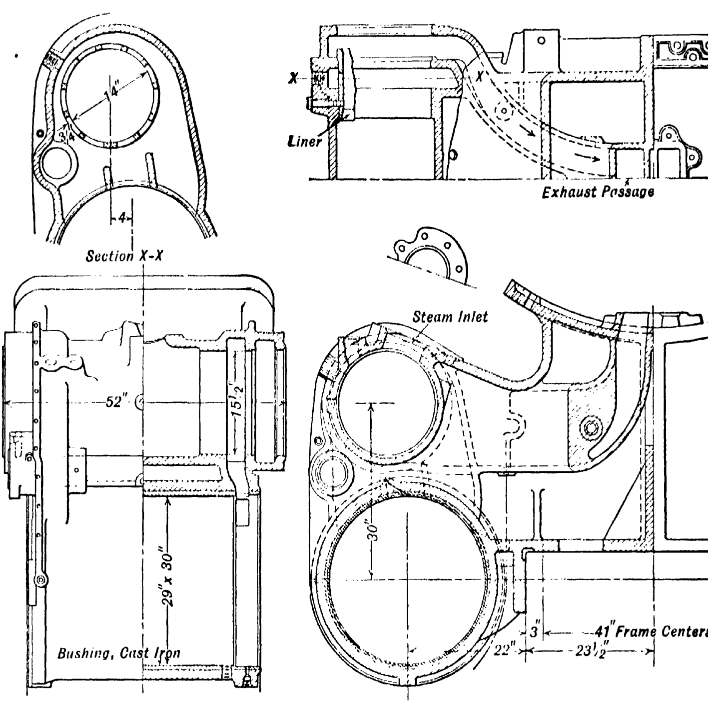

FIG. 258. — Portion of a 29 in. by 30 in. (4-8-2) Locomotive Cylinder Casting.

the saddle is sometimes made separate and each cylinder bolted to it. The frame is bolted to the cylinder casting as shown. Steam from the boiler is carried to the steam inlet by the steam pipe, and then passes to the valve chest through a cored steam passage. The exhaust steam passes from the cylinder, through the exhaust passage, to a **nozzle** in the smoke box located directly underneath the stack.

The locomotive does not require a foundation. The wheels and the weights they carry take the place of the flywheel. The wheels, axles, connecting rods, parallel rods, piston rods, crossheads and frame compose the **running gear.** The wheels attached to the connecting rod are known as **drivers,** and are connected by **side rods,** so that they will revolve together. Each driver is counter-balanced by a crescent-shaped weight located opposite each crank pin, to balance the revolving and recipro-

cating parts and thus produce smooth running. The wheel frames are made of cast iron or steel, are forced onto the shaft and have hardened steel tires which are shrunk on.

The valve may be of the balanced, flat, or piston type. A piston-type valve that has a packing ring, to prevent leakage past the valve, is shown in Fig. 259. The dimensions give some idea of the size of the valve. Most locomotive valves, instead of having an exhaust lap, have an exhaust

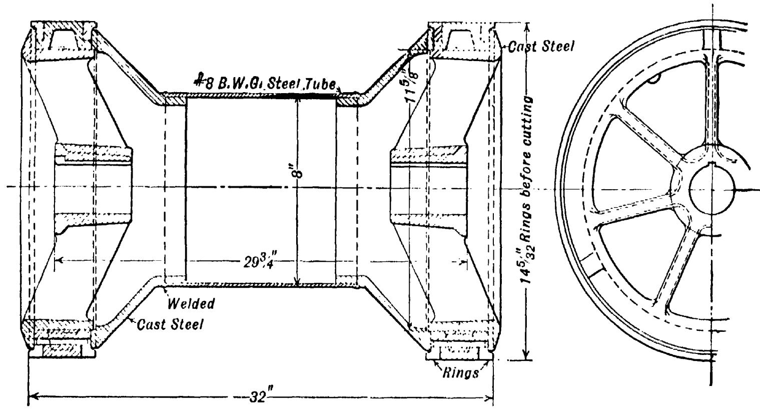

Fig. 259. — Piston Valve for (4-8-2) Locomotive.

clearance of from $\frac{1}{8}$ to $\frac{3}{16}$ inch. This reduces the back pressure on the engine, since the exhaust port is thus held open for a longer period.

The locomotive valve gear should be easy to operate, and should be so made that the engine can be reversed at will. The principal locomotive valve gears used in American practice, together with typical makes, are given below:

1. Link motions	Stephenson Gooch	
2. Radial valve gears	Walschaert Baker Young Southern	Modern Types
	Joy Marshall Hackworth	Old types

351. Stephenson Link Motion. — In Fig. 260, e represents the position of the center of the eccentric, with reference to the crank, for a plain slide valve having a fixed eccentric, which is connected to the valve rod without an intervening rocker and which has an outside admission. Such an

engine can be reversed by turning the eccentric around the shaft until the center of the eccentric is at e', directly across the shaft from e. To reverse the engine by this method would not be practical for locomotive, marine, hoisting, traction and rolling mill engines, which have to be reversed frequently.

The Stephenson link motion, Fig. 261, has two fixed eccentrics, A and B, each having its own eccentric rod. The eccentrics are located essentially as the eccentrics would be in Fig. 260. The ends of the eccentric rods are connected by a **slotted link,** suspended by a **link hanger rod** from a **bell crank,** which is pivoted to a shaft carried by the frame and is operated by the reversing lever. A **saddle block** attached to the back of the link supports a pin to which the lower end of the hanger rod is attached. A slide block which fits the slot in the link, and is free to move in it, is connected to the end of the valve rod. The slide block and valve rod remain stationary during movement of the link by the bell crank.

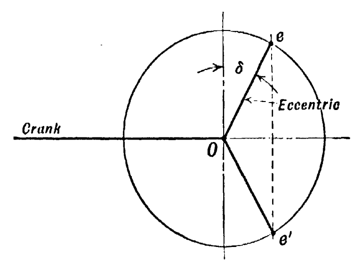

FIG. 260. — Diagram showing Positions of Eccentric when Reversing Rotation of Engine.

The eccentric rods may be open, as in Fig. 261, or crossed. For an outside admission valve and no reversing rocker arm, the link motion is open

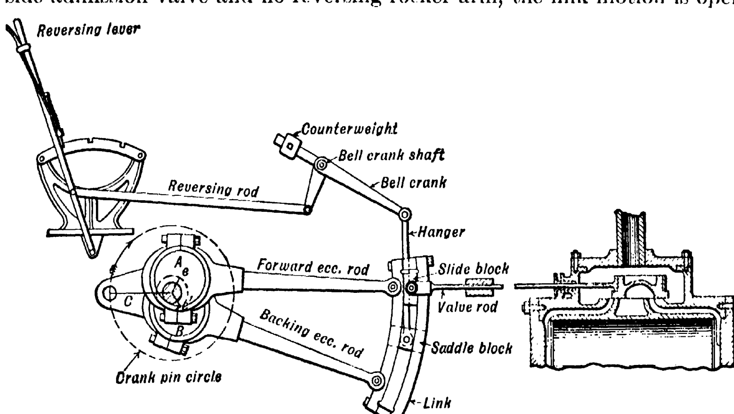

FIG. 261. — Stephenson Link Motion.

rod construction, if the rods are open when the crank is in the dead-center position on the side of the shaft away from the valve. Under the same conditions, the valve gear is of the crossed rod construction if the rods

are crossed when the crank is in the dead-center position on the side away from the valve gear.

With the reversing lever in the position shown, the valve is moved by the forward eccentric A. In this case eccentric B has small effect upon the motion of the valve.

When the link is raised to bring the valve rod in line with the backing eccentric rod, the motion of the valve comes mainly from the eccentric B, and the engine runs in a counter-clockwise direction. In this case the eccentric A has little effect on the motion of the valve.

With the block midway between two ends of the link, the valve is acted upon equally by both eccentrics, one tending to produce clockwise rotation, and the other, counter-clockwise rotation; therefore, the engine will not run in either direction.

The valve mechanism is said to be in **mid-gear** when the block is in the middle of the link, and to be in **full-gear** when the block is at its extreme position near the end of the link. There are two full-gear positions, one called **full-gear forward** and the other, **full-gear reverse.**

With the link in full-gear position, the full steam pressure acts upon the piston for nearly the entire stroke, and the latest cut-off is obtained. This is the condition at starting a locomotive pulling a load. As the link is brought toward mid-gear position, the cut-off becomes earlier.

For the Stephenson link motion, with open-rod construction, the lead increases as the link is moved from full-gear to mid-gear, while with the crossed-rod construction the lead decreases as the link is moved from full-gear to mid-gear. For use on locomotives, it is desirable to have the lead increase from full-gear to mid-gear, in order to give more cushioning effect as the engine speeds up.

352. The Walschaert Radial Valve Gear. — The Walschaert valve gear was invented in 1844, but has not been used extensively in this country until recently. The arrangement of the parts of the Walschaert gear, as applied to a locomotive, is shown in Fig. 262. With the mechanism in the position shown, the valve would admit steam to the head end of the cylinder.

The eccentricity is obtained by an **eccentric crank** keyed to the main crank pin. The point E is approximately 90 degrees from the dead-center position of the crank. An eccentric rod connects the eccentric crank to the **reversing link,** which is slotted and is supported, at its middle point, on the frame of the engine by a **trunnion.** The eccentric rod is attached to an arm that projects from the bottom of the link, in order to bring the eccentric rod as near the horizontal as possible. This arm is shaped to correct the distorted motion of the valve, caused by the angularity of the eccentric rod. A **radius rod** connects the link with the **lap-and-lead lever,** or **combination lever,** and thus transmits the movement of the link to the valve.

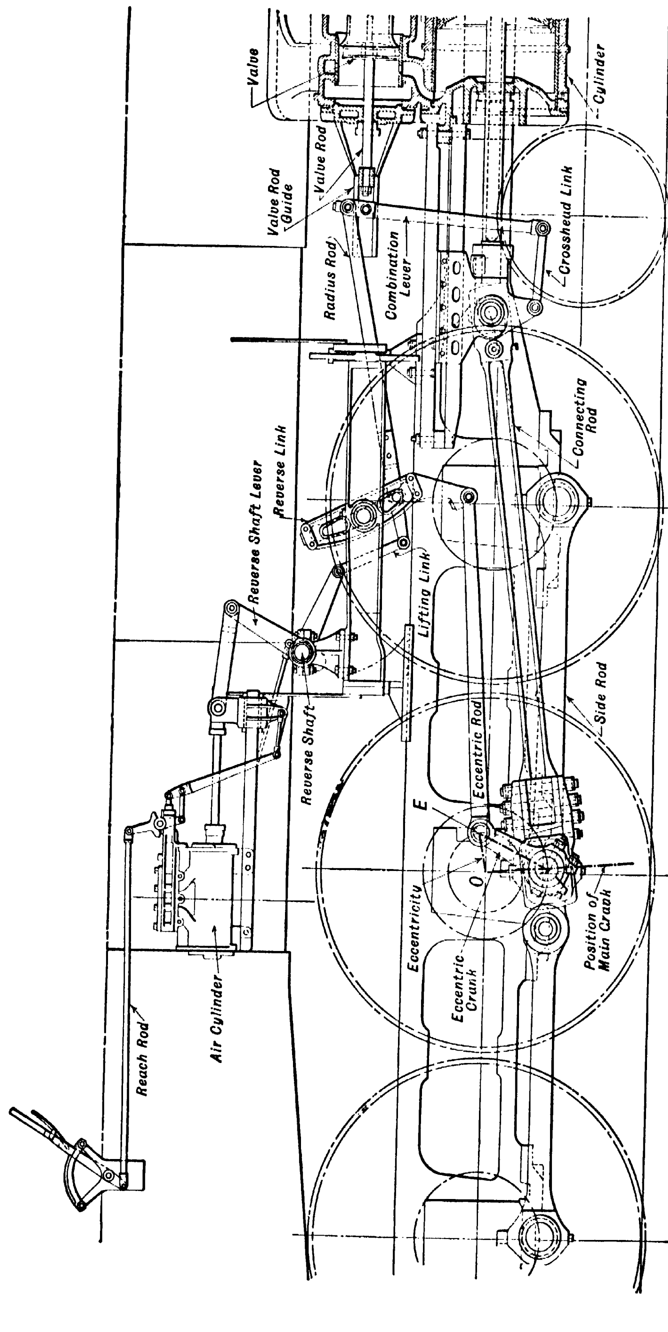

FIG. 262. — Arrangement of Walschaert Valve Gear for Chicago, Burlington and Quincy Pacific (4-6-2) Locomotive.

The reversing link is curved to a radius equal to the length of the radius rod, in order to keep the lead constant.

To the link end of the radius rod, a **movable block** is fastened. It is moved up and down in the slotted link by a bell crank pivoted on the frame. One arm of the bell crank is attached to the radius rod by a lifting arm and the other arm is attached to the reach rod, operated from the cab. When the block is above the link fulcrum the engine runs backward, and when the block is below the fulcrum the engine runs forward.

The point of cut-off is latest with the block at either end of the reverse link, for in this position the valve travel is a maximum. As the block is moved toward the center of the link, the cut-off occurs earlier, and the valve travel is decreased. The travel of the valve, in any case, is less than twice the eccentricity, because the reversing link and lap-and-lead lever act as a rocker arm and reduce the travel.

The valve motion is the resultant of the motion given by the reverse link, and the lap-and-lead lever. With the engine on dead center, the eccentric crank is so set that the link is in its middle position. If the radius rod were attached directly to the valve stem, the valve would be in its mid-position with the block at either end of the link, and steam would not be admitted. *To have the valve displaced from its mid-position by an amount equal to the lap plus the lead, and thus have steam admitted to the cylinder, the lap-and-lead lever is used.* One end of the lever is pivoted to the valve stem, and the other end is attached to a link, which is fastened to an arm on the crosshead. The outer end of the valve rod slides in a guide attached to the engine frame. The lap-and-lead lever is so proportioned that the valve would be moved a distance equal to twice the sum of the lap plus the lead, when the point of connection to the radius rod is kept a stationary fulcrum and the piston moved a distance equal to the stroke. The lap-and-lead lever gives the effect of the angle of advance.

For a valve having *outside admission,* the valve stem is connected to the lap-and-lead lever at a point above that at which the radius rod is connected. For an *inside admission* valve, the valve stem is connected to the lap-and-lead lever at a point below that at which the radius rod is connected, Fig. 262.

For outside admission, with the block in the lower half of the link and in forward gear, the eccentric leads the main pin. With the block in the upper half of the link and in forward gear, the eccentric follows the crank. For an inside admission valve, the position of the eccentric for each of the above cases is reversed.

The Walschaert and similar gears are used extensively, for the following reasons:

1. With the weight, size, and power of modern locomotives, it is difficult to design a satisfactory Stephenson link motion that works

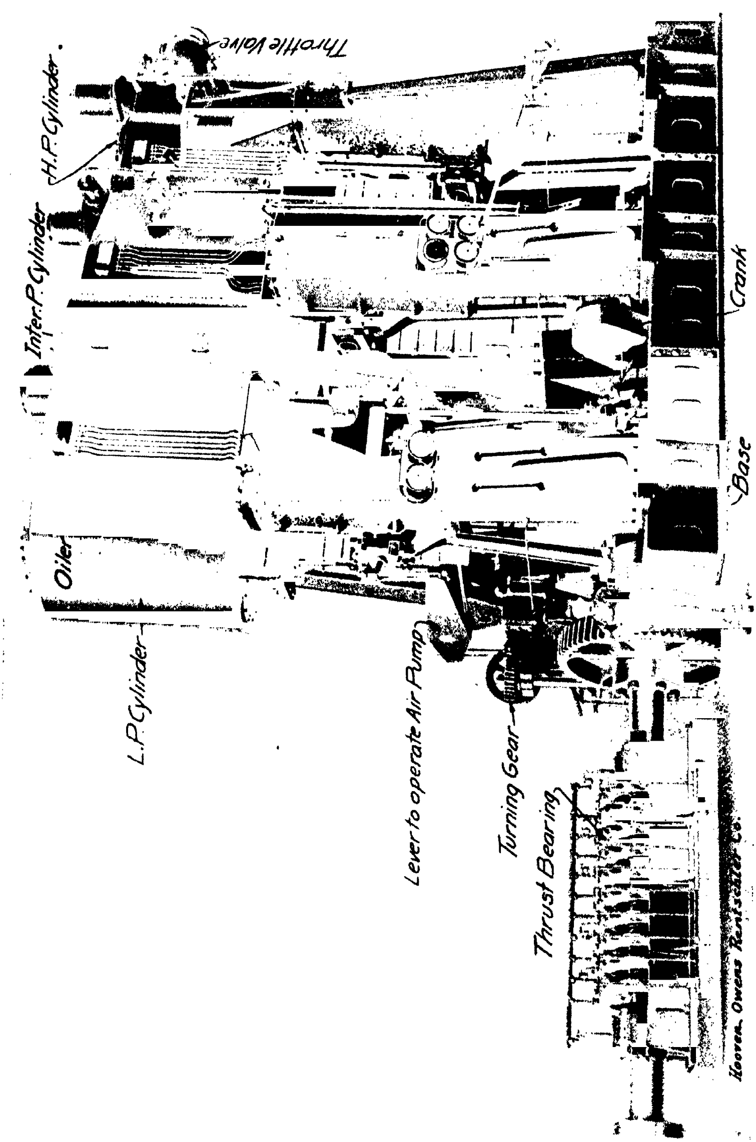

FIG. 263. — 2500 Horsepower Triple-expansion Marine Engine.

between the sides of the frame. The radial gears, being **outside** the frame, overcome this difficulty.

2. The straps and eccentrics of the Stephenson link motion are heavy and wide. They are located underneath the frame where they are subject to dust, and rapid wear results. This, together with the wear in the rockers and transmission bars, results in a large amount of lost motion. The radial gears do not have eccentrics, and as only hardened pins are used, they are more easily kept in repair.
3. The position of the radial gears, outside the frame, makes them more accessible for proper lubrication.
4. The frame can be made stronger by bracing, because the valve gear is not inside the frame.

The locomotive does not require a governor. The speed is regulated by changing the pressure of the steam admitted to the cylinder by the throttle valve, or by changing the point of cut-off by means of the valve gear.

353. Marine Steam Engine. — A marine engine of the latest type is shown in Fig. 263. These engines are seldom made with a single cylinder, but are made compound and multiple-expansion, the steam passing in succession from one cylinder to one or more larger cylinders. The most common types are compound, triple-expansion, and quadruple-expansion engines, and are made in sizes up to 6000 horsepower.

The frame consists of inverted columns resting on a heavy bed-plate, which has six cross girders by which the main bearings are supported. Two-part babbitted bearings are used, with the lower part of the bearing housing cast as a part of the bed-plate.

The valves are balanced piston and balanced flat valves, actuated by a Stephenson link motion which permits reversing the direction of rotation of the engine. A small steam engine attached to the main frame operates the valve gear when reversing. The valves are attached to balancing pistons, located in cylinders above the valves, to counterbalance the weight and inertia of the valves and valve gear, when conditions require. Crossed eccentric rods are generally used to permit stopping of the engine by setting the link at mid-gear. With open rods and the link at mid-gear, the engine might not stop. All parts of the valve gear are made heavy, and the moving parts are often babbitted.

A governor is not required, as the resistance of the propellor increases, with the velocity, sufficiently to control the speed. For low speeds the engine is governed by means of the throttle valve.

The thrust, or push, of the propellor is taken by a **thrust bearing** and transferred to the frames of the ship. This bearing is at the left of the

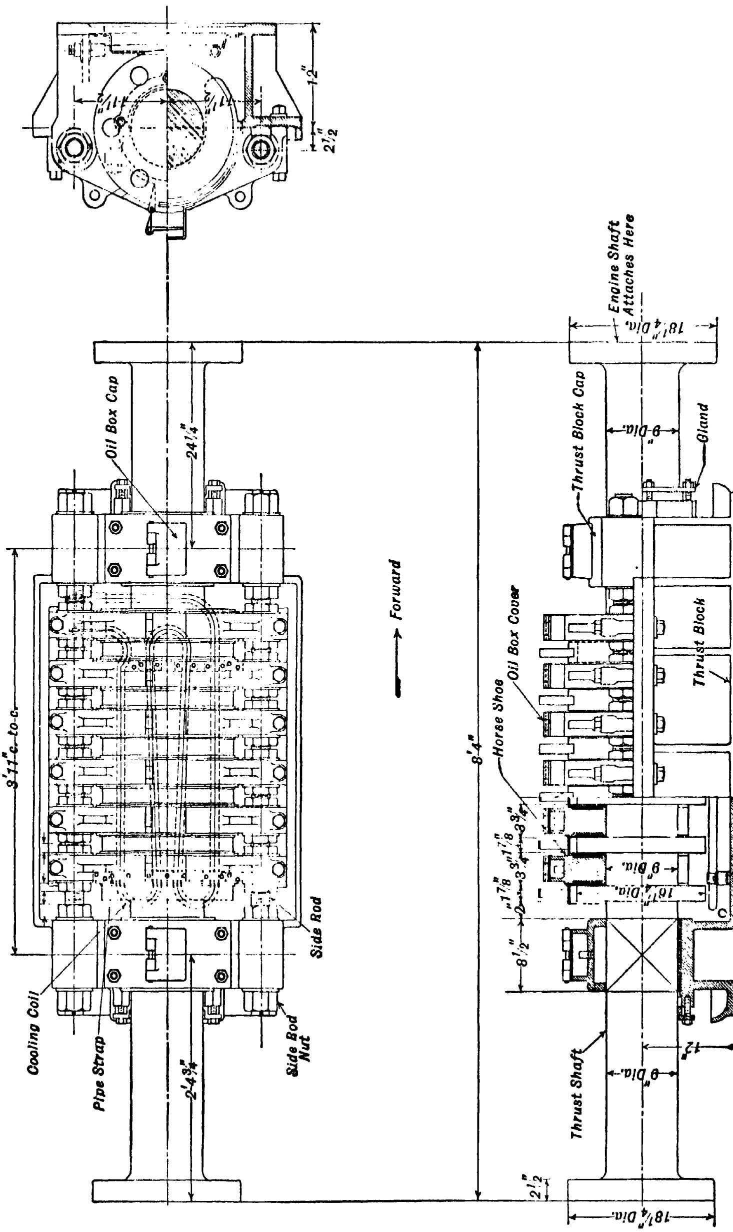

FIG. 264. — Horseshoe Type of Thrust Bearing for 850 hp. Marine Engine.

illustration and is shown more in detail in Fig. 264. It is a modern type of **horseshoe-collar thrust bearing,** having a series of rings or **collars** made on the shaft and a series of separate thrust yokes, which fit between the shaft collars. These thrust yokes are provided with ears, or lugs, which

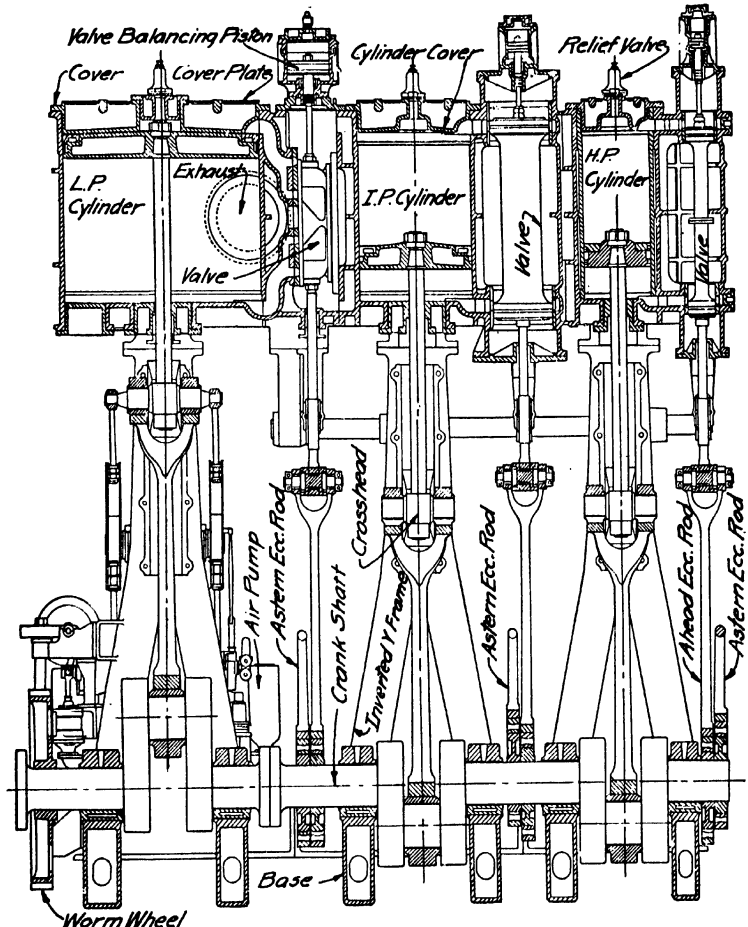

FIG. 265. — Vertical Section thru Engine shown in Fig. 263.

bear against nuts on side rods attached to the bearing casing. This arrangement permits each thrust yoke to be adjusted separately for wear.

The thrust from the propellor is transferred from the faces of the shaft rings to the thrust yokes, thence through the lugs and nuts to the side rods, which transmit the thrust to the bearing casing bolted to the frames of the ship. Each thrust yoke has an oil chamber at the top, from which

the oil is siphoned by wicking into the oil pipe, and in addition the lower part of the bearing housing is made rectangular and filled with oil, into which the shaft collars dip.

The **turning gear** used to turn the main shaft, when the main engine is not operating, is shown to the left of the engine. It consists of a large **worm wheel** on the main shaft, geared to a **worm**. On some engines the worm is operated by a small steam engine, and on others by hand; a lever with **pawl** and **ratchet** being used in the latter case, to operate the worm.

An **air pump** is attached to the side of the low-pressure cylinder frame, and is driven by the low-pressure crosshead through a lever connection. It is used to maintain the vacuum on the engine when operating condensing.

The eccentric is made of cast iron and keyed to the shaft, and the strap is also made of cast iron and babbitted. The eccentric rod, to which the forked valve-rod end is bolted, is of forged steel.

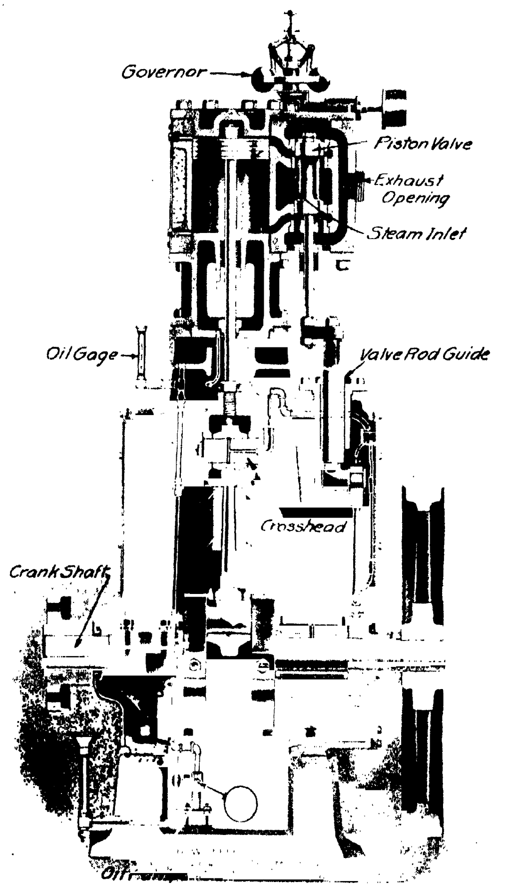

FIG. 266. — Small Vertical Steam Engine.

The crosshead may be of the single or double slipper type, and the guides are formed as a part of the main frame. The sectional view, Fig. 265, illustrates the arrangement of the various operating parts.

354. Small Vertical Slide-valve Engine. — This type of engine is used to drive stokers, fans, and other auxiliaries. The engine shown in Fig. 266 has an enclosed frame with balanced piston valve. The frame is of the *A*-type, completely enclosed. The guides are formed as a part of the main frame, which is supported by a bed-plate that makes a reservoir for oil. Between the cylinder and the top of the frame, is a distance piece which prevents water that leaks past the piston-rod stuffing box from getting into the enclosed frame. All moving parts are completely enclosed.

355. Special Forms of Slide Valves. — In addition to the slide valves already described, the following types are used on slide-valve engines:

1. Double and multi-ported.
2. Riding cut-off.

356. Double and Multi-ported Valves. — The plain slide valve gives a restricted port opening at the start of the stroke. To overcome this, and to reduce the movement of the valve necessary for a given port opening, **multi-ported** valves are used. Figure 267 shows a **double-ported** valve used on marine engines. The valve is surrounded by steam, and has two steam chambers that run clear through the valve and are filled with live steam. The valve seat has two ports for each end, which merge into a passage connecting with the cylinder. The two ports on each end are covered with valve feet which are duplicates and, as the valve moves, both ports are opened at the same time, thus increasing the size of the passage for steam for any movement of the valve.

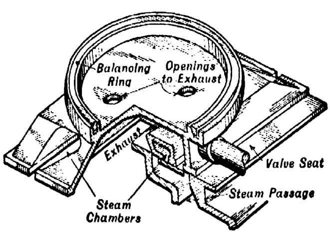

Fig. 267. — Marine Double-ported Slide Valves.

A double-ported valve used on locomotives, and known as the **Allen trick valve,** is shown in Fig. 268. It resembles a "*D*" slide valve but has a steam passage cored in it. The valve is so proportioned that when it is in the position shown steam will enter the cylinder, past the outer edge of the valve and also from the crank end through the cored passage. This gives a more rapid opening and closing of the port.

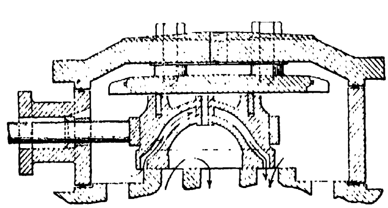

Fig. 268. — Allen-Richardson Balanced Slide Valve — Double-ported.

A multi-ported valve known as the **gridiron** valve is used on multi-valve engines. It consists of a flat plate having a number of parallel rectangular passages through it. The valve seat has an equal number

of parallel rectangular ports, which are simultaneously opened and closed by the action of the valve. When this type of valve is used, a separate valve or valves must be used for the inlet and exhaust.

357. Riding Cut-off Valves. — It was shown in Art. 345, page 327 that, with a single valve controlling all events of the engine, a change in the cut-off, by changing the angle of advance, also changed the point at which compression, release, and admission occurred, an earlier cut-off being accompanied by an earlier compression.

High compression on a slow-running engine is not desirable, and to provide a means of changing the cut-off without altering the point at which release, compression, and admission occur, a **riding cut-off valve** is used.

A simple form of this valve is shown in Fig. 269. A **main valve** which controls the point of admission, compression and release, slides upon the valve seat. It is essentially a plain slide valve having steam passages near the ends. Steam enters the cylinder through these passages, instead of past the outside edge of the valve, and a riding cut-off valve slides upon the back of the main valve and controls the point of cut-off. It is a flat valve with a projection to which the riding cut-off valve stem is attached.

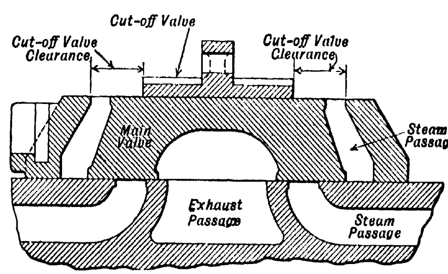

FIG. 269. — Riding Cut-off Valve.

The main and riding cut-off valves are controlled by separate eccentrics. The eccentric for the main valve is fixed and set to bring cut-off at about three-quarter stroke. The riding valve eccentric is loose upon the shaft, and is generally controlled by a shaft governor which controls the speed by changing the point of cut-off. The riding valve and its eccentric are so designed and adjusted that at admission the cut-off valve is not obstructing the port but at cut-off the block is moved to cover the port before the main valve has reached its cut-off position. Having closed the valve port for either end, the riding valve must keep it closed until the main port is closed by the main valve, or a second admission of live steam will occur.

When the riding valve is located centrally on the back of the main valve, it fails to cover the ports in the main valve by an amount C, called **cut-off valve clearance,** or **negative lap.** The riding valve must be dis-

354. Small Vertical Slide-valve Engine. — This type of engine is used to drive stokers, fans, and other auxiliaries. The engine shown in Fig. 266 has an enclosed frame with balanced piston valve. The frame is of the A-type, completely enclosed. The guides are formed as a part of the main frame, which is supported by a bed-plate that makes a reservoir for oil. Between the cylinder and the top of the frame, is a distance piece which prevents water that leaks past the piston-rod stuffing box from getting into the enclosed frame. All moving parts are completely enclosed.

355. Special Forms of Slide Valves. — In addition to the slide valves already described, the following types are used on slide-valve engines:

1. Double and multi-ported.
2. Riding cut-off.

356. Double and Multi-ported Valves. — The plain slide valve gives a restricted port opening at the start of the stroke. To overcome this, and to reduce the movement of the valve necessary for a given port opening, **multi-ported** valves are used. Figure 267 shows a **double-ported** valve used on marine engines. The valve is surrounded by steam, and has two steam chambers that run clear through the valve and are filled with live steam. The valve seat has two ports for each end, which merge into a passage connecting with the cylinder. The two ports on each end are covered with valve feet which are duplicates and, as the valve moves, both ports are opened at the same time, thus increasing the size of the passage for steam for any movement of the valve.

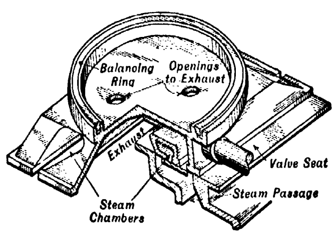

FIG. 267. — Marine Double-ported Slide Valves.

A double-ported valve used on locomotives, and known as the **Allen trick valve,** is shown in Fig. 268. It resembles a "D" slide valve but has a steam passage cored in it. The valve is so proportioned that when it is in the position shown steam will enter the cylinder, past the outer edge of the valve and also from the crank end through the cored passage. This gives a more rapid opening and closing of the port.

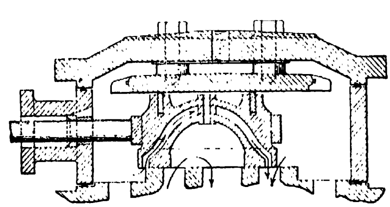

FIG. 268. — Allen-Richardson Balanced Slide Valve — Double-ported.

A multi-ported valve known as the **gridiron** valve is used on multi-valve engines. It consists of a flat plate having a number of parallel rectangular passages through it. The valve seat has an equal number

of parallel rectangular ports, which are simultaneously opened and closed by the action of the valve. When this type of valve is used, a separate valve or valves must be used for the inlet and exhaust.

357. Riding Cut-off Valves. — It was shown in Art. 345, page 327 that, with a single valve controlling all events of the engine, a change in the cut-off, by changing the angle of advance, also changed the point at which compression, release, and admission occurred, an earlier cut-off being accompanied by an earlier compression.

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A simple form of this valve is shown in Fig. 269. A **main valve** which controls the point of admission, compression and release, slides upon the valve seat. It is essentially a plain slide valve having steam passages near the ends. Steam enters the cylinder through these passages, instead of past the outside edge of the valve, and a riding cut-off valve slides upon the back of the main valve and controls the point of cut-off. It is a flat valve with a projection to which the riding cut-off valve stem is attached.

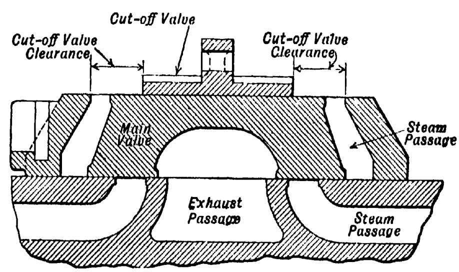

FIG. 269. — Riding Cut-off Valve.

The main and riding cut-off valves are controlled by separate eccentrics. The eccentric for the main valve is fixed and set to bring cut-off at about three-quarter stroke. The riding valve eccentric is loose upon the shaft, and is generally controlled by a shaft governor which controls the speed by changing the point of cut-off. The riding valve and its eccentric are so designed and adjusted that at admission the cut-off valve is not obstructing the port but at cut-off the block is moved to cover the port before the main valve has reached its cut-off position. Having closed the valve port for either end, the riding valve must keep it closed until the main port is closed by the main valve, or a second admission of live steam will occur.

When the riding valve is located centrally on the back of the main valve, it fails to cover the ports in the main valve by an amount C, called **cut-off valve clearance,** or **negative lap.** The riding valve must be dis-

of steam, the cut-off at each end of the cylinder should be equal, and for smooth running the lead at each end should be equal, in order to give equal cushioning effect at each end of the stroke. Valves are therefore set for **equal leads** or equal **cut-offs.** With the ordinary construction of slide valves, the steam laps on each end are equal. When so made, a valve set for equal leads will not give equal cut-offs, because of the angularity of the connecting rod, and for the same reason, a valve set for equal cut-offs will not give equal leads. Valves are ordinarily set for equal leads.

In setting a slide valve, the laps and eccentricity are fixed in amount, and adjustments that can be made are:

1. Moving the valve on its stem.
2. Changing the position of the eccentric.

Moving the valve on the stem has the effect of changing the laps. When it is moved bodily along on the seat, the result is as though the steam lap were increased and the exhaust lap were decreased for the end toward which it is moved, and vice versa for the opposite end. Hence, changing the length of the valve rod increases the lead on one end of the valve and decreases it on the other. Lengthening the valve rod increases the lead at the crank end and decreases it at the head end.

Shifting the position of the eccentric around on the shaft increases or decreases both leads, depending upon whether the angle of advance is increased or decreased.

360. Setting the Crank on Dead Center. — In setting valves it is necessary to place the crank on dead center. This must be accurately

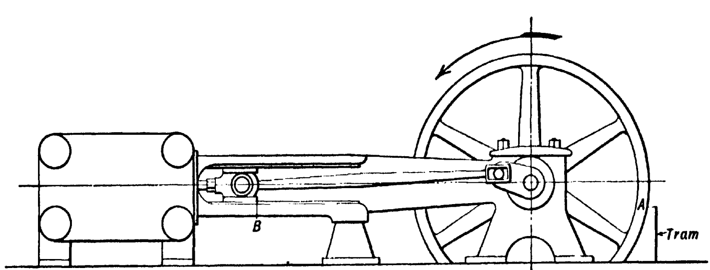

FIG. 273. — Method of Marking Guide.

done, because when near dead center the crank and eccentric move through a considerable angle for a slight movement of the piston and crosshead. If the dead center is not carefully found, the valve will not be set in its proper position relative to the piston.

The **trammel method** is generally used to place the engine on center. The **tram** is a rod pointed at both ends with one end bent to form a right angle.

The flywheel is turned until the crosshead is within 2 or 3 inches of the end of the stroke, and a mark is made on a chalked surface on both guide and crosshead, as shown at *B*, Fig. 273 Whenever these marks are brought together, the crosshead will be in its original position. The straight end of the tram is then placed on a fixed mark on the floor or frame, and a mark is made with the bent end of the tram on the rim of the flywheel, as shown at *A*, Fig. 273. The crank pin is then turned past the dead-center position until the marks on crosshead and guide again coincide and a second mark is made with the tram on the rim of the flywheel

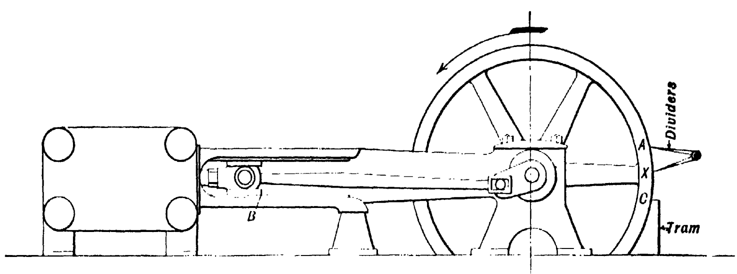

FIG. 274. — Method of Using Tram and Dividers.

as shown at *C* in Fig. 274. The crank is now as far past dead center as it originally was ahead. A point *X* is located mid-way between *A* and *C*, and a mark is made at *X*. With the tram resting on the permanent mark on the floor, the engine is turned until the bent end of the tram rests squarely on the line at *X*. The engine is now exactly on dead center. The dead center for each end should be found in the above manner. The engine should always be turned, to take up any lost motion, in the same direction as when running, and thus have the same brasses in contact with the crank pin; otherwise the setting may not be correct.

361. Setting a Plain Slide Valve for Equal Leads. — The methods employed to set valves are:

1. By removing the valve-chest cover and locating the valve by measurement.

2. By taking indicator diagrams and adjusting the valve gear until the desired diagrams are obtained.

362. Setting a Plain Slide Valve for Equal Leads, by Measurement. — The valve is first adjusted to give equal travel each side of mid-position, or to have equal preliminary leads. The setting by the second method is made as follows:

1. With the crank on dead center, the angle between crank and eccentric is adjusted until the valve opens the port leading to the cylinder, by a slight amount. The width of the opening should be measured and

recorded as a preliminary lead on that end. Suppose, for example, that it is $\frac{1}{8}$ inch. The engine should now be placed on the opposite dead center, and the port opening on that end measured and recorded as the preliminary lead on that end. Suppose it to be $\frac{1}{16}$ inch; there is, then, $\frac{1}{16}$ inch difference in the lead on the two ends. To make the leads equal in amount, the valve must be moved on its stem, or the length of the valve rod changed by an amount equal to half the difference. The valve should be moved toward the port having the larger opening.

2. With the leads equal, but not necessarily correct in amount, the required lead is given by placing the crank on head-end dead center and, after loosening the eccentric, moving it around the shaft, in the direction the engine is to run, until the desired lead is obtained. The engine is then turned to the other dead center and the lead on the crank end checked. The leads should now be nearly equal. The head-end lead is generally made slightly less than the crank-end lead, because of the distortion caused by the angularity of the connecting rod.

363. Setting a Plain Slide Valve for Equal Cut-offs. — 1. The valve is first set to travel equal distances each side of mid-position, as explained under setting for equal leads. 2. The eccentric is then adjusted to give cut-off when the piston has moved equal distances from dead center for each end. This is done by marking the limits of the stroke on the guide and setting the crosshead at the percentage of stroke at which cut-off is to occur. The eccentric is moved on the shaft in the direction the engine is to run, until it can be seen that the valve is just closing the steam port for the end from which the piston is moving. The eccentric is then fastened on the shaft and the engine turned over until the crosshead has moved the same distance from the other dead center and the valve should be just closing the port. If the setting is not correct the difference in measurements should be halved and a correction made for one-half by moving the eccentric on the shaft, and for the other half by moving the valve on the stem. This operation should be repeated until the required setting is attained.

Setting a Slide Valve by Means of a Tram. — Valves are often set by means of a tram and tram-marks on the valve stem. By this method the valve need not be seen and the steam chest cover is not removed. The tram-marks on the stem are obtained as follows: a center punch mark is made on some fixed part of the engine, and the straight part of the tram placed in it. The valve is then moved until its edge is at the head-end edge of port and a punch mark made on the valve stem into which the bent end of tram will fall. A similar punch mark is made on the valve rod when the edge of the valve is at the edge of the crank-end port. Knowing the position of the valve as indicated by these marks, the valve can be quickly adjusted.

For valve setting by indicator, see Art. 395, page 395.

364. Setting Special Types of Slide Valves. — For valves controlled by shaft governors, the length of the valve stem and position of the eccentric are found as for the plain slide-valve engine. The eccentric position is determined by turning the whole governor wheel and eccentric until the desired setting is obtained. This is done with the governor giving latest cut-off. For an engine having reversing devices, the setting is made as in the case of valves driven by a fixed eccentric, except that the valve must be set for both full-gear forward and reverse.

For riding cut-off valves, the main valve is set for equal leads, exactly as for a single slide valve. The riding valve is set to give equal cut-offs at some point of the stroke. When rocker arms are used, care must be taken to turn the eccentric in a direction to open the port with the crank pin on dead center. The cut-off is generally equalized at the middle of the governor range.

On a shaft-governor engine, the weights should be blocked out against the action of the springs, to the position for earliest cut-off, thus making sure that the governor can cut-off sufficiently near the beginning of the stroke to control the engine.

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Bulletin No. 1018, AMERICAN LOCOMOTIVE CO.
Practical Marine Engineering, DYSON.

REVIEW QUESTIONS AND PROBLEMS

1. Name the parts of the eccentric connection to the valve rod, and define eccentricity.

2. Describe the operation of a "D" slide-valve engine.

3. What type of governor is used with the plain "D" slide valve? How does it control the speed?

4. Define: (*a*) valve travel, (*b*) mid-position of valve, (*c*) lap, (*d*) lead, (*e*) port opening.

5. How is the displacement, of the piston and valve respectively, stated?

6. Why is a valve diagram used? Describe the Zeuner diagram and state its fundamental principle.

7. A $6\frac{1}{2}$ in. by 10 in. engine with a connecting rod 30 in. long has a valve travel of 3 in., a steam lap = $\frac{3}{4}$ in., a lead = $\frac{1}{8}$ in., an exhaust lap = $\frac{3}{16}$ in. Use a 6 in. circle and find, by the Zeuner and Bilgram diagram, the distance from the end of the stroke at which the following occur for each end of the cylinder: (*a*) cut-off, (*b*) release, (*c*) compression.

8. Given: admission at $\frac{1}{120}$ before start of stroke, cut-off at $\frac{3}{4}$ stroke, valve travel,

$4\frac{1}{4}$ in. Find the steam lap, lead and angle of advance, using the Zeuner and Bilgram diagram. Neglect the effect of the angularity of the connecting rod.

9. What is meant by an automatic high-speed engine? What method of governing is used on such an engine?

10. Describe the operation of the Rites inertia governor.

11. Describe the construction of the Walschaert valve gear.

12. Describe the construction of a marine thrust bearing of the collar type. Why is it used?

13. Describe the construction and operation of a Meyer valve.

14. Find the radius of suspension and the coördinates of the point of suspension, for a swinging eccentric on a 7 in. by $10\frac{1}{2}$ in. engine having a steam lap = $\frac{5}{8}$ in.; cut-offs at 1, $3\frac{1}{16}$ and 7 in. on stroke; lead at the corresponding cut-offs, −0.035, 0.00, 0.055 in. Neglect the effect of the angularity of the connecting rod.

15. Explain the meaning of the term "relative valve circle."

16. State the methods used to set a slide valve.

17. Explain the trammel method of finding dead center.

18. Explain the method of setting a slide valve for equal leads.

19. In setting a slide valve on a 6 in. by 6 in. engine, it was found upon trial that with the proper lead on one end, the opposite end gave $\frac{1}{8}$ in. too great a lead. State what should be done to obtain correct setting and give reasons for your answer.

20. A slide-valve engine has a reverse rocker between the valve and the eccentric rod. What should be the location of the eccentric center with respect to the crank, to have the engine run over?

21. What is the advantage of using trams in setting a slide valve?

22. State the procedure in setting (*a*) a slide valve controlled by a shaft governor. (*b*) a riding cut-off valve.

CHAPTER XVIII

MULTI-VALVE ENGINES

365. Foreword. — It has been shown in Chapter XVII that, when a single valve controls all events of the engine, the opening of the steam port is restricted at admission. This causes throttling, or a drop in pressure, between the valve chest and the cylinder. Engines having single valves have large clearances because of the long steam passages, and the exhaust steam in passing from the cylinder cools the steam passages and cylinder walls. When live steam again comes in contact with the cooled walls, some of it is condensed and, therefore, more steam is required to perform a given amount of work.

By the use of engines having four valves — two for admission and two for exhaust — the steam passages may be shortened and the clearance

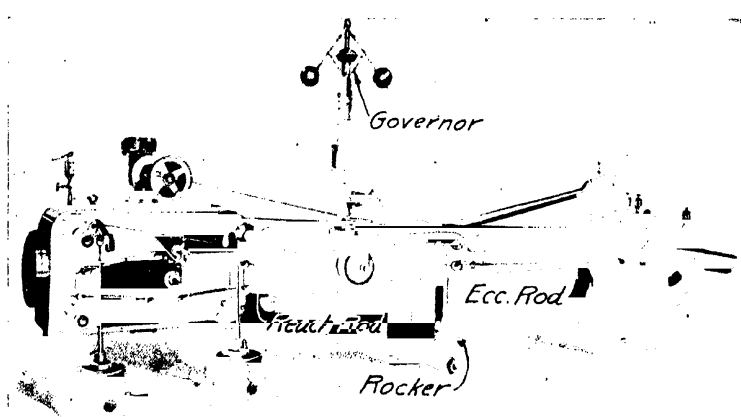


FIG. 275. — Corliss Engine—Valve Gear Side.

space reduced, because the valves may be located close to the ends of the cylinder. The use of separate valves for admission and cut-off permits the separate adjustment of each valve, and also reduces the initial condensation of steam.

366. Corliss Engine. — All parts of this engine, Fig. 275, with the exception of the valves, valve gear and method of governing, were described in Chapter XVI. The Corliss valve gear, invented by Geo. H. Corliss in

1850, has been principally instrumental in reducing the amount of steam, per horsepower, used by large reciprocating engines; and this has been the predominating type of valve gear used, since that time, on important stationary engines in the United States, England, and France. It consists essentially of cylindrical valves, which are given a semi-rotary motion by a releasing mechanism, under control of a governor. Valve gears using cylindrical valves and a non-releasing gear, which gives the same effect as

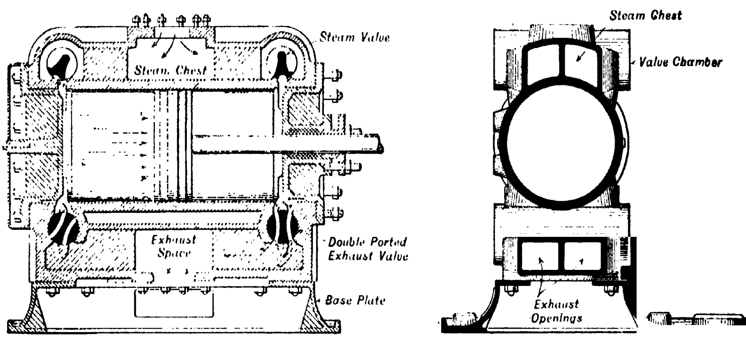


FIG. 276. — Corliss Cylinder showing Location of Valves.

the original Corliss releasing gear, are also termed Corliss gears. The Corliss engine is essentially a slow-speed engine, since satisfactory operation of the ordinary Corliss valve gear limits the speed to 100 or 125 revolutions per minute.

367. Classification of Corliss Valve Gears. — Corliss valve gears may be classified as:

1. Standard releasing.
2. Long range cut-off.
3. High speed.
4. Non-releasing, or positively operated.

368. Corliss Valves. — These valves are cylindrical, cast-iron pieces, extending across the cylinder, and are usually located at the four corners of the cylinder casting, Fig. 276. Engines which have large cylinders often have the valves in the cylinder heads, to simplify the cylinder casting. The **steam valves** of the horizontal engine, Fig. 277, are located at the top and control admission and cut-off; the **exhaust valves,** Fig. 277, are located at the bottom and control exhaust and compression. The valves may be single-ported, double-ported, or multi-ported. By the use of more than one port, the movement of the valve necessary for a given port opening is decreased.

The ends of the valves are made cylindrical, to form a surface which supports the valve. The middle portion of the steam valve is cut away,

leaving a narrow valve face, which permits the valve to be moved more easily. Steam pressure acts on the valve and holds it on its seat. The

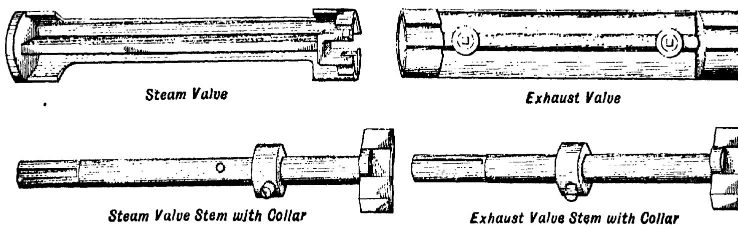
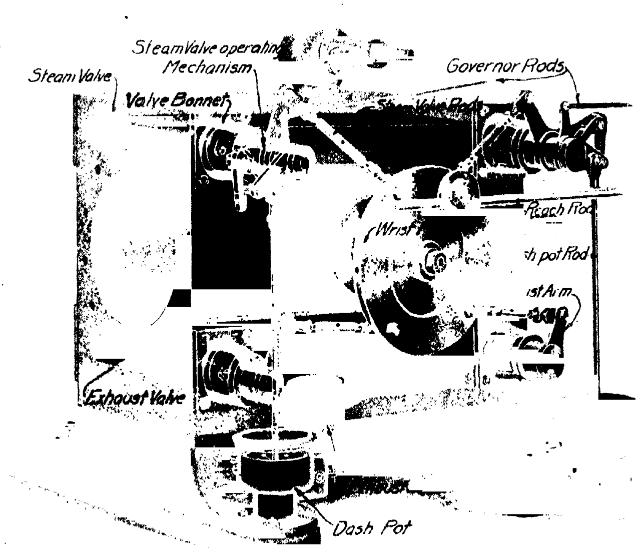


FIG. 277. — Steam and Exhaust Valves of a Corliss Engine.

exhaust valve is cut away only enough to form the steam passage. The valves oscillate back and forth, to open and close the ports.

The valve stem is made of brass and has a rectangular end which fits into a corresponding slot cut in the end of the valve. It extends through a



Courtesy Vacuum Oil Co.

FIG. 278. — Standard Releasing Corliss Valve Gear.

stuffing box in the front bonnet, which carries a bracket to support the valve operating mechanism, and to form a bearing for the outer end of the valve stem. A collar on the valve stem takes the end thrust of the valve.

369. Standard Releasing Corliss Valve Gear. — In the standard valve-gear mechanism, shown in Fig. 278, the valves are moved by arms and rods attached to the **wrist plate,** which is pivoted on the cylinder casting.

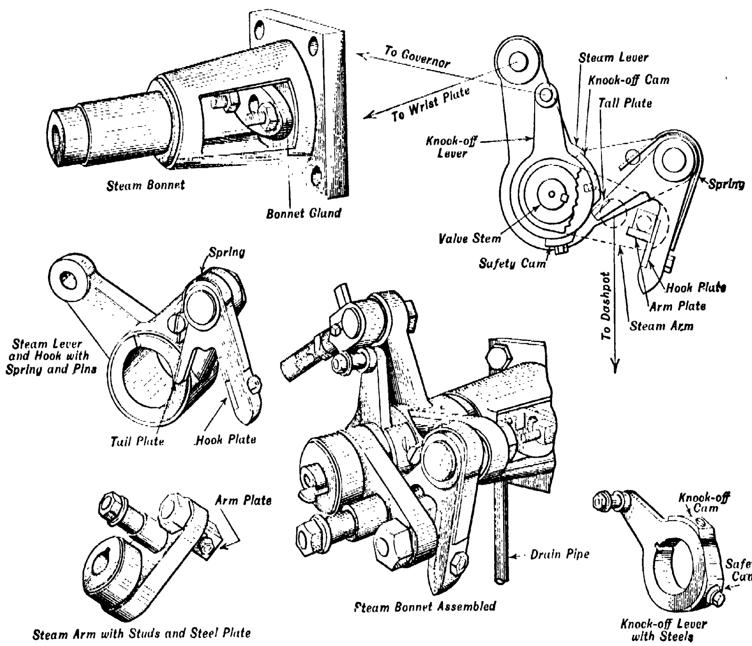


FIG. 279. — Corliss Detaching Valve Gear.

The **steam rods** are attached to the wrist plate in such a manner that the valves, while opening, move rapidly and cover a large proportion of their total travel. The wrist plate receives its motion from the eccentric through the eccentric rod and the **reach rod.** A **rocker arm** is placed between the reach rod and eccentric rod to reduce the length of the eccentric rod and to increase the movement of the wrist plate for a given movement of the eccentric.

The exhaust valves are attached permanently to the wrist plate by **exhaust arms** and adjustable **exhaust-valve rods.** They are thus positively operated and receive, from the wrist plate, an oscillating motion, which is small while the valves are closed.

A **trip mechanism,** shown assembled and disassembled in Fig. 279, controls the operation of the steam valves, since they are not attached directly to the wrist plate. They are opened by movement of the wrist plate, released by the **knock-off-cam,** and closed by the **dash pot.** The valve stem is keyed to the steam-valve arm, to which the **dash-pot rod** is attached.

Mounted to turn on the **bonnet bracket,** in which the valve stem turns, are the **bell-crank lever** and the **governor cam-plate lever.** These levers are not connected directly to the valve stem. One arm of the bell-crank lever is attached to the wrist plate by an adjustable steam-valve rod; to the other arm of the bell-crank lever is pivoted the **steam-valve latch,** or hook, which turns against the action of a flat spring. The **cam plate,** which lies in the plane of the valve hook, carries an arm, connected to the governor by a **governor rod** and **lever.** With the wrist plate, Fig. 278, at the extreme left of its travel, the latch is forced in by the flat spring and engages the **latch plate,** or block, on the head-end steam-valve arm. As the eccentric travels to the right, it rotates the bell crank and lifts the steam-valve arm, which rotates the steam valve on its seat and opens the steam port, and at the same time lifts the dash pot. The movement of the valve is rapid at this point, to prevent throttling of the steam pressure. The valve is held open by the latch, until the arm of the latch, which is not engaged with the latch plate, meets the cam on the knock-off lever. The cam pressing on this arm then forces the latch to release the latch plate, and the dash pot

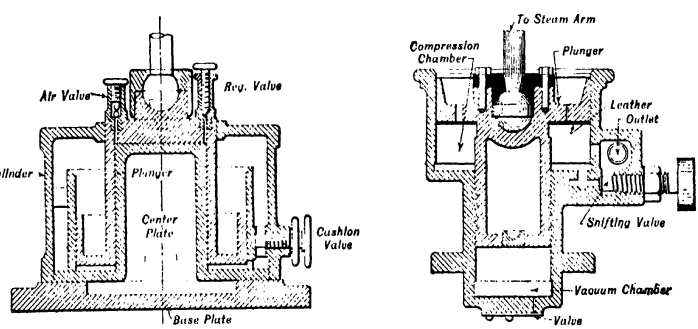


FIG. 280. — Dash Pots.

closes the valve. The cam plate is held stationary, for any given speed, by the governor. The same action is then repeated by the crank-end mechanism.

For the cylinder in Fig. 276, steam is flowing into the head end and exhaust steam out of the crank end, through the double-ported valves.

370. Dash pot. — The dash pot is required to perform two functions: *it must exert a downward pull on the valve, and it must be self-cushioning.*

There are numerous types and arrangements of dash pots, two of which are shown in Fig. 280, with the various parts named. The outer casting, or cylinder, is generally fastened to the bed-plate by bolts. The **plunger** fits into the cylinder, and at its upper end is attached to the steam valve arm by the dash-pot rod. The joint between the plunger and the dash-pot rod is a **ball and socket joint,** adjustable for wear. The lower part of the cast-

ing into which the lower end of the plunger fits forms the **vacuum chamber,** at the bottom of which is a small hole covered by a **flat spring** or **snifting valve.** The upper part of the cylinder and casting is enlarged to form an **air pocket** for cushioning the plunger. The air pocket is connected to a **cushion valve** which regulates the amount of cushioning.

The plunger is raised by the latch, and a partial vacuum is thus formed below it. When raised, the plunger has the pressure of the atmosphere acting on its upper surface and a lower pressure, depending upon the degree of vacuum, acting on the bottom part of the plunger. The unbalanced pressure closes the valve suddenly when the latch is released by the knock-off cam. As the plunger descends, air is caught in the **cushion space** and escapes slowly through the **cushion-regulating valve.** This brings the dash pot to rest quietly, and without shock. Any air in the vacuum chamber is forced out through the opening at the bottom, and the vacuum is thus maintained. In some recent types of dash pots, a spring is used to cushion the plunger.

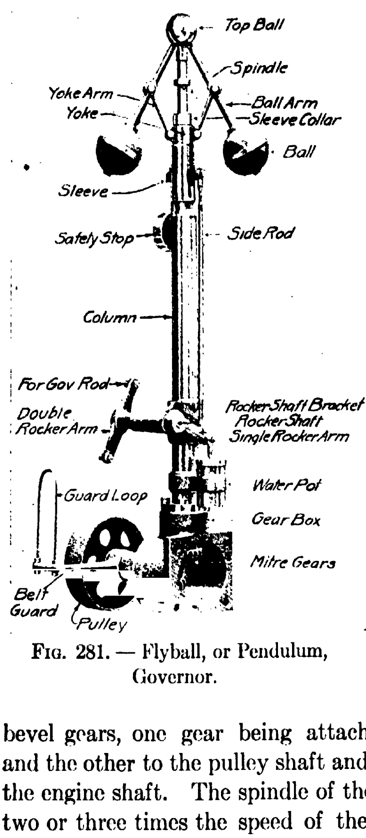


FIG. 281. — Flyball, or Pendulum, Governor.

371. Governor. — The governor used on most Corliss engines having a releasing valve gear, is the **Watt,** or **pendulum governor,** Fig. 281. The governor balls are pivoted to the top of a spindle by ball arms. The spindle is driven through bevel gears, one gear being attached to the lower end of the spindle and the other to the pulley shaft and pulley which is driven by a belt from the engine shaft. The spindle of the latest types of governors revolves at two or three times the speed of the engine shaft. This permits smaller weights to be used and increases the sensitiveness of the governor. The spindle revolves in a vertical column attached to the engine frame, and, at its upper end, is surrounded by a movable sleeve to which a yoke connected to the ball arms is attached. The sleeve is connected by a **side rod** to a **double rocker arm** to which the governor rods are connected. It rises and falls with change in speed and thus determines the position of the knock-off cams and consequently the point of cut-off. A movable

collar is located on the spindle to prevent the balls from rising too high, and a dash pot is attached to the side rod to prevent sudden fluctuations in the action of the governor.

A recent form of Corliss governor, made by the Allis Chalmers Co., has heavy weights acting against a spring. The weights and springs replace the balls and the force of gravity of the Watt governor. This makes a more compact, but a more complicated form of governor. All rotating parts are enclosed in a stationary case which is filled with oil.

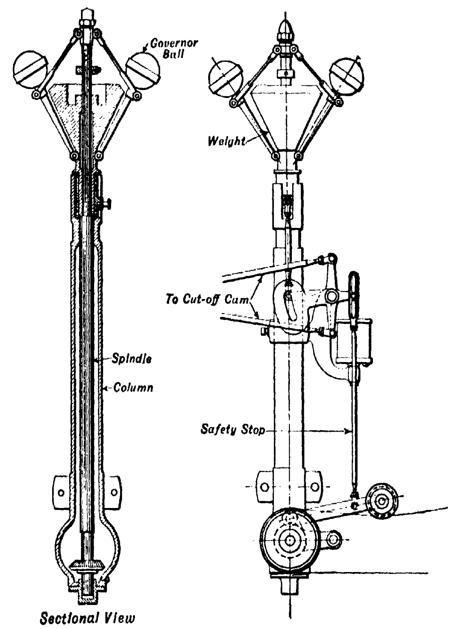


Fig. 282. — Loaded Pendulum Governor.

Loaded governors, Fig. 282, that have a heavy weight mounted on the spindle sleeve, are used extensively. This type of governor is more powerful than one without the weight, and it can be operated at a lower speed without loss of sensitiveness.

A **safety stop** is ordinarily attached to the governor. Should the governor balls fall to their lowest position for any reason, the safety stop

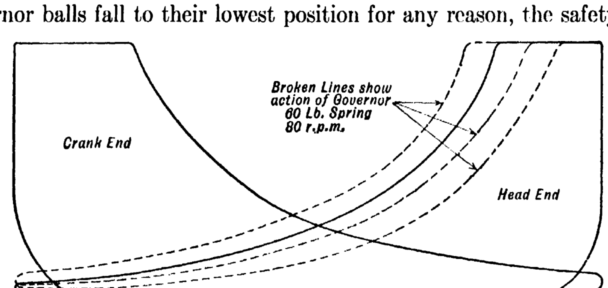


Fig. 283. — Indicator Card from 16″ × 38″, Single Eccentric Corliss Engine.

moves the cam-plate lever into such a position that the **safety cams** located on the cam-plate lever will prevent the latch plate from picking up the valve arm, and steam will not be admitted to the cylinder.

Typical Corliss diagrams, showing the effect of the operation of the Corliss governor, are shown in Fig. 283. The diagrams show the rapid operation of the valves at admission and at cut-off.

372. Limitation in Range of Cut-off for Single Eccentric Corliss Valve Gear. — Referring to Fig. 284; with the crank on head-end dead center, the eccentric will be at $1a$. The latest point of cut-off will occur when the crank is at 2, because the eccentric has then advanced to its extreme position $2a$, and any further movement will cause the eccentric to travel to the left. If the latch arm has not been released by the governor cam before the eccentric is at $2a$, it will not be released, and the valve will not cut-off under governor action, but will be closed by the eccentric in the same manner that an ordinary slide valve is closed. This limits the cut-off by the governor in a single eccentric engine, with the angle of advance as given, to about 0.4 of the stroke.

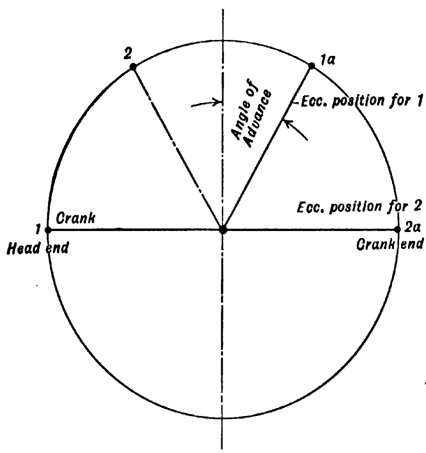


FIG. 284. — Illustrating Limitation in Range of Cut-off with Single Eccentric.

By retarding the eccentric to less than 90 degrees, cut-off by the governor can be made later. This, however, destroys the advantage of the Corliss gear, namely, quick opening of valves, because with the crank at dead center the eccentric is also nearer dead center and the speed of opening of the valves is reduced. Retarding the eccentric in this manner also affects the operation of the exhaust valves, since they are operated by the same eccentric.

By using two eccentrics, one for the admission valves and one for the exhaust valves, the cut-off eccentric may be set to give any cut-off up to seven-eighths stroke, and the exhaust valve eccentric may be set to give the proper compression and release.

373. Nordberg Long-range Corliss Valve Gear. — The essential difference between this gear, Fig. 285, and the single eccentric Corliss valve gear is that the latch is positively thrown in and out of engagement with the **drop arm,** by means of a lever which terminates in a roller resting in an **oscillating cam** attached to the valve arm. *This cam has two concentric circular slots joined by a transition slot. When the roller passes from one slot to the other, the latch is released and cut-off occurs.* The cam is oscil-

lated by the **cut-off rod** attached to the governor, Fig. 286, and receives its motion from a separate cut-off eccentric, the throw of which corresponds to the length of each circular slot in the cam. The position of the cam and cut-off is determined by the governor and the cut-off eccentric. This gear permits higher speeds, because only one eccentric is used, and gives a range of cut-off up to 0.8 the stroke. The movement of the valve, given by this gear, is shown by the valve ellipse, Fig. 287.

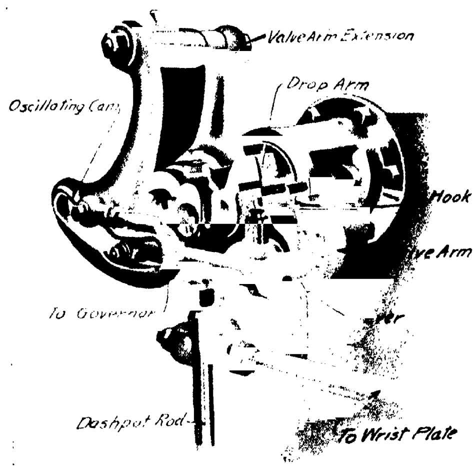


FIG. 285. — Nordberg Long Range Valve Gear.

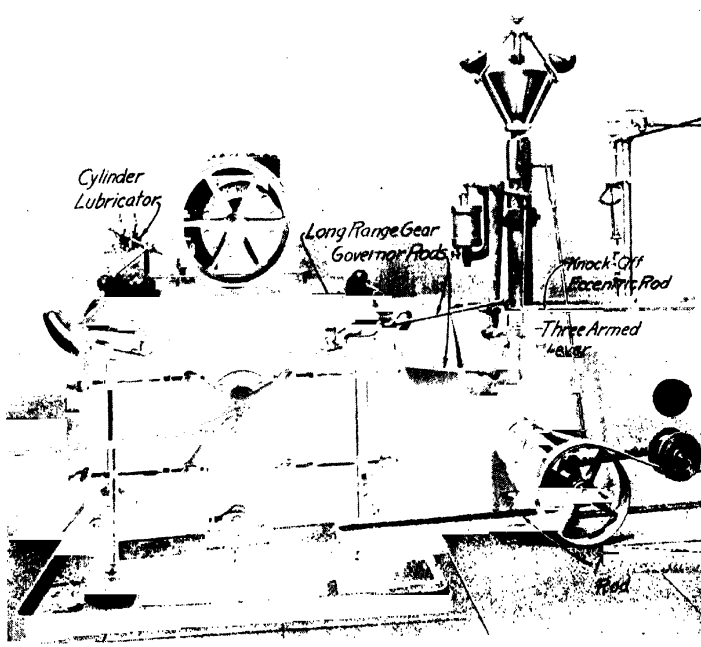


FIG. 286. — Nordberg Engine with Long Range Valve Gear.

374. Nordberg High-speed Corliss Valve Gear. — The ordinary releasing Corliss gear is limited in speed because of the inability of the releasing mechanism to operate satisfactorily under high speeds. The Nordberg high-speed valve gear is adapted to speeds as high as 250 r.p.m. The essential difference between this type and the long-range cut-off gear is that the trip mechanism is symmetrical with respect to the central plane, and does not have heavy overhanging parts. The valve stem and arm are supported by two bearings which support the stem on each side of the latch. The valves are made four-ported, to reduce the travel of the valve mechanism, and the dash pot is mounted on top of the valve bonnet.

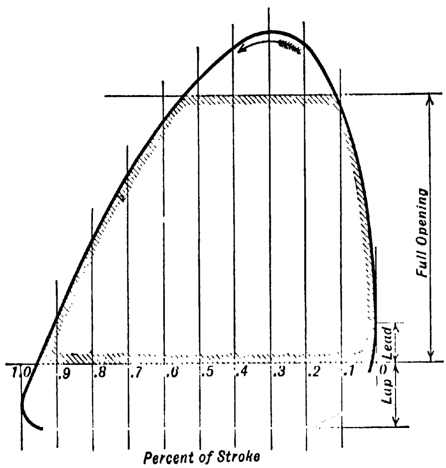


FIG. 287. — Valve Ellipse for Admission Valve operated by Nordberg Long Range Valve Gear.

375. Positively Operated Corliss Gears. — Engines having this type of gear are sometimes called **four-valve engines.** The typical Corliss valve motion is attained by using a system of small links, proportioned to give the valve the desired motion. The pins on which the links oscillate are

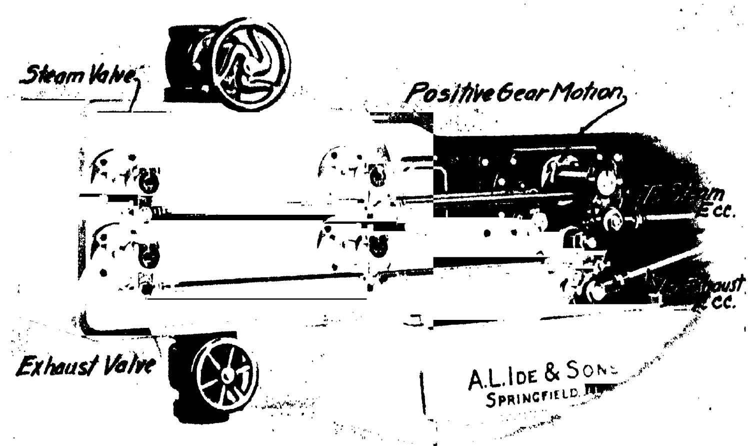


FIG. 288. — Ideal Non-releasing Corliss Valve Gear.

made of hardened steel. In the **Ideal non-releasing gear,** shown in Fig. 288, the exhaust valves are driven by one eccentric and the steam valves by another. The steam-valve eccentric rod is connected to a **positive gear motion** attached to the side of the frame, and is connected separately to each steam valve. The link motion is shown in Fig. 289. The links move in an oil bath and are so proportioned that the valves remain at rest while closed, during their unbalanced period, for approximately one-half a revolution of the engine. The valves are opened and closed rapidly while balanced; that is, with the steam pressure nearly the same on both sides. The elements of this motion are shown in Fig. 290, where lines represent the various parts of the gear. With the eccentric at 1, the various links are at the corresponding points also marked 1. The **rocker arm** is moved by the eccentric, and drives the valve linkages for each end. To each end of the **rocker arm** is pivoted a lever on which is formed a **cylindrical tail rod**

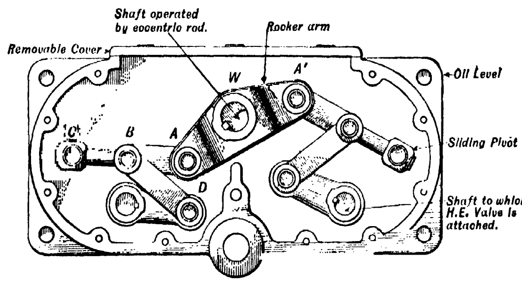


FIG. 289. — Ideal Link Motion to operate Valves.

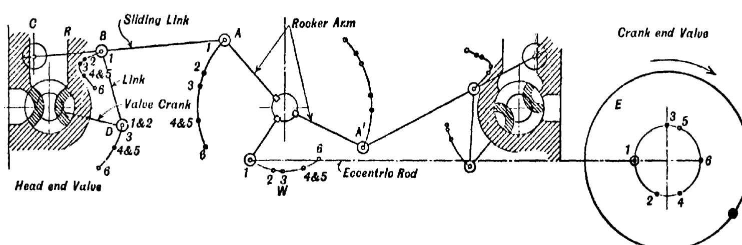


FIG. 290. — Line Diagram of Ideal Corliss Valve Gear.

that slips through a **trunnioned guide,** which serves as a fulcrum. At a proper point on the sliding lever, a link, which drives the valve arm, is pivoted. The corresponding points of the linkage and eccentric positions are marked. This gear can be operated at speeds greatly in excess of those used with the releasing gear.

376. Corliss Valve Setting. — The principles that apply to the setting of the plain " D " slide-valve engine are applicable to setting Corliss valves. The valves of four-valve engines may be set by essentially the same method as that used for Corliss valves. The method used in setting Corliss valves is as follows:

1. The head- and crank-end dead-center positions are determined, as explained in Art. 360, page 352.

2. The bonnets covering the rear end of the valves are removed and the marks on each valve and seat verified. The mark on the end of each valve locates the position of the working, or opening edge, of the valve, and a corresponding mark on each seat locates the opening edge of the port.

3. The length of the eccentric rod is adjusted to make the rocker swing through equal angles on each side of its vertical position. This may be tested by using a **plumb bob** and marking each position on the floor. The dash-pot rods should be short enough to escape being caught and bent when turning the engine " over " in making this test.

4. The reach rod is next adjusted, if it is adjustable, to make the wrist plate swing through equal angles each side of its vertical position, when the engine is turned over. Marks are usually found on the hub, supporting the wrist plate. The outside marks locate the extreme positions of the travel of the wrist plate, and a mark midway between the outside marks locates the vertical position. On the hub of the wrist plate a single line will be found. When this line coincides with the central line on the hub, the wrist plate is **vertical,** or **central.** This position corresponds to that of the slide valve in mid-position, and it is the position of the wrist plate when measuring the laps.

5. The steam arms are hooked up, the wrist plate placed central, and the steam valves given the proper amount of lap, by adjusting the lengths of the steam links. The best method of measuring the amount of lap is by using a pair of dividers and a scale. The exhaust links are next adjusted to be line-and-line, or to have a small clearance. See Table 26 for proper laps and leads.

TABLE 26. — LAPS AND LEADS FOR CORLISS VALVES

Diameter of Cylinder, In.	Lap of Steam Valve, In.	Lap of Exhaust Valve, In.	Lead of Steam Valve, In.
8	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{32}$
10	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{32}$
12	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{32}$
14	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{32}$
16	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{32}$
18	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{32}$
20	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{32}$
22	$\frac{5}{16}$	$\frac{3}{16}$	$\frac{3}{64}$
24	$\frac{5}{16}$	$\frac{3}{16}$	$\frac{3}{64}$
26	$\frac{5}{16}$	$\frac{3}{16}$	$\frac{3}{64}$
28	$\frac{5}{16}$	$\frac{3}{16}$	$\frac{3}{64}$
30	$\frac{5}{16}$	$\frac{3}{16}$	$\frac{3}{64}$
32	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{16}$
34	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{16}$
36	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{16}$

6. The crank is now placed on head-end dead center, the steam valves hooked up, and, with the wrist plate connected to the reach rod, the eccentric is moved *around the shaft in the direction the engine is to run,* until the steam valve nearest the piston shows the desired lead. The eccentric is next fastened on the shaft and the engine shaft turned in the direction it is to run, to the other dead center and the lead noted. If the leads are not the same, the connection between the wrist plate and eccentric is shortened or lengthened slightly or the length of the steam link changed to give the desired lead. This affects the steam lap, but not seriously.

7. The length of the dash-pot rod is adjusted to give the proper latch clearance, by turning the eccentric; first, to one extreme of its travel and lengthening or shortening the dash-pot rod to have the latch block clear the latch by about $\frac{1}{16}$ inch, and then, to the other extreme and making the same adjustment.

8. The governor rods are adjusted to give equal cut-off at each end, as follows: Block the governor halfway between its up and down positions and see if the rocker arm to which the governor rods are attached is about at right angles to a line midway between the governor rods. Place the piston at one-quarter stroke and raise the governor slowly until the valve is released by the latch Block the governor in this position and turn engine over, to see if cut-off occurs at same point for the other end. If it does not, shorten or lengthen the governor rod for that end to make the latch release the valve. Block the governor in a new position and see if cut-off is equal on each end. *A few trials may be needed to strike an average.* If desired, the governor may be blocked in its running position.

9. The governor balls are now placed in their lowest position and safety stop or cam adjusted, as the case may be, to prevent the latch from picking up the hook.

10. The correctness of the settings is tested by means of an indicator, Art. 395, page 395. For double eccentric Corliss engines, the eccentric is placed central with no lap on any of the valves, and the steam valves set as for a single eccentric. The exhaust eccentric is set by placing the piston at seven-eighths stroke and advancing the eccentric until the exhaust valve, at the end the piston is approaching, is just beginning to close. The piston on the opposite end is next placed at seven-eighths stroke, and the process repeated for the other exhaust valve.

If there is any doubt as to the direction in which to move the eccentrics, remember that the *eccentric should always be set to have steam valve opening when crank starts from dead center, in direction engine is to run,* and that the exhaust valve at same end should be closing.

377. Poppet-valve Engine. — The poppet-valve engine is used to a greater extent in Europe than in the United States. The Lentz poppet-valve engine was developed in 1899. Since that time many improvements

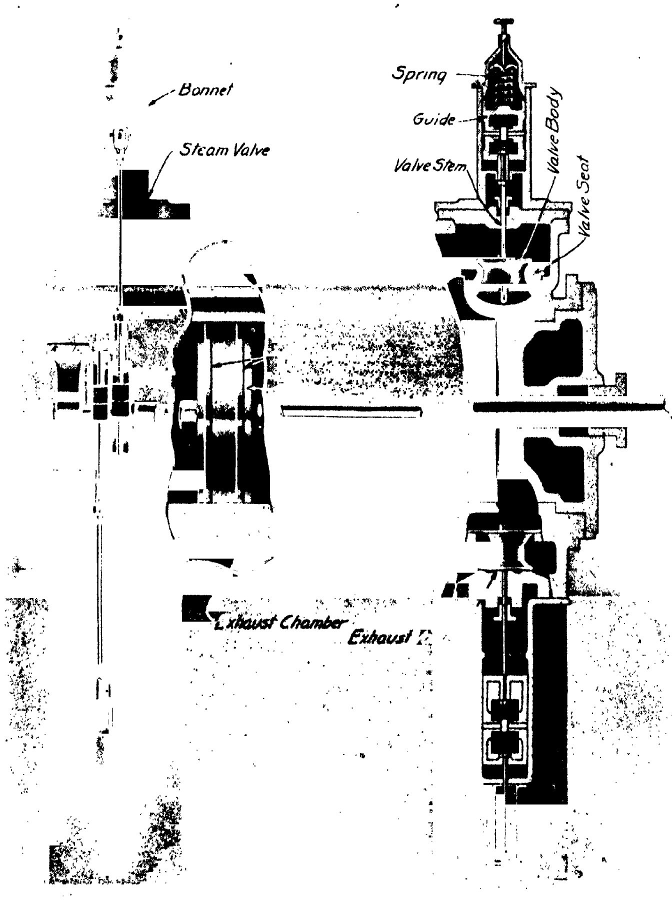


FIG. 291. — Poppett Valve Engine Cylinder.

have been made and it is now used extensively. *Poppet valves are adapted to high speeds and can be used with superheated steam.*

A common type of poppet-valve engine cylinder, shown in Fig. 291, has two steam and two exhaust valves, each valve consisting of two cast-

iron disks joined by a cylindrical body. The faces of the upper and lower disks rest on seats in the valve cage when the valve is closed, and the form

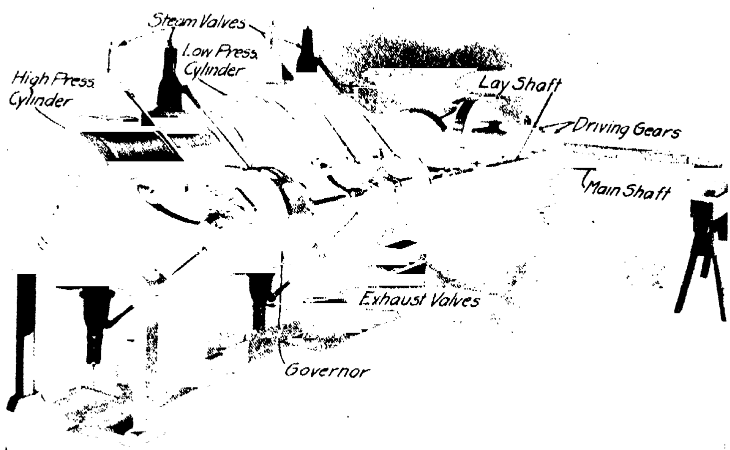


FIG. 292. — Lentz Poppet-Valve Compound Engine.

of the valve makes it nearly balanced. The valve remains stationary when closed. The valve stem passes through a stuffing box in the valve cage, and is attached to a valve-stem guide which operates against a spring at the top of the bonnet. The valve cage and bonnet are removable and are bolted to the top and bottom at each end of the cylinder. Steam passages connect the valve seat and the cylinder.

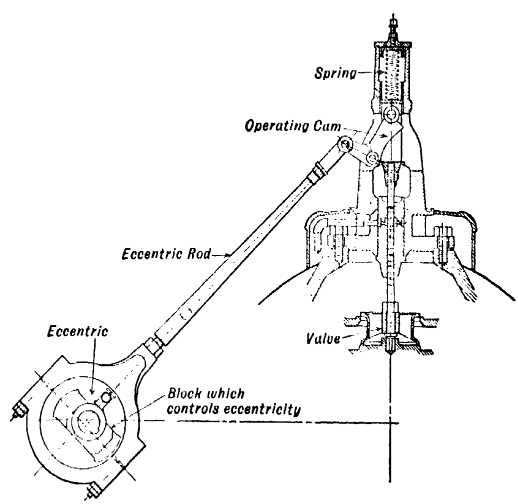


FIG. 293. — Admission Valve Gear of Lentz Engine.

The Lentz poppet-valve gear is shown in Fig. 292, and the admission valve gear in more detail in Fig. 293. The valve stem extends from the valve through a bearing that serves as a stuffing box. In order to prevent loss of steam.

the valve stem has grooves cut around its circumference where it slides in the bearing.

A spring which closes the valve presses on the upper end of the valve stem and bears against an **oscillating cam.** The cam is moved by an eccentric located on a **lay shaft,** which runs the length of the engine and is driven by bevel gears from the main shaft.* Each valve has a separate eccentric. The exhaust-valve eccentrics are fixed on the lay shaft to open

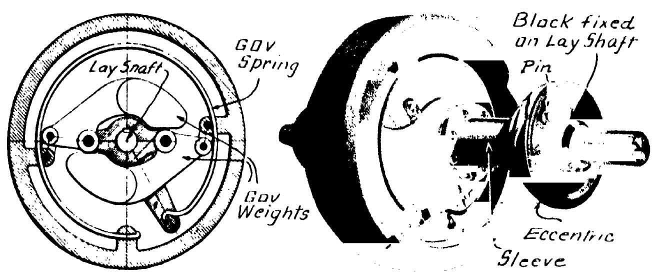


FIG. 294. — Governor of Lentz Poppet Valve Engine

and close the exhaust valves at the proper time. The steam-valve eccentrics are slotted, and are under the control of the governor, which regulates the speed by changing the point of cut-off.

The small centrifugal governor, Fig. 294, is mounted on a sleeve which rotates on the lay shaft. The governor sleeve carries a pin, connected to the eccentric in such a way that it moves the eccentric on a block, which is fixed on the governor shaft, thus changing the eccentricity, and consequently the length of time the valve remains open.

A poppet-valve engine made by the Nordberg Engine Co. differs from that previously described, in that the cylinder proper is a straight barrel without valve passages. The valves, together with the steam and exhaust passages, are located in the cylinder heads, which are bolted to each end of the barrel. The valves are positively opened and closed, without the aid of a spring, by a cam which acts on a roll plate pivoted to the bonnet and attached to the valve stem. The operation of this engine is similar to that of the Lentz engine.

378. Unaflow Engine.† — In the engines previously described, steam enters the cylinder at each end, and is exhausted at the same end at which it enters. The walls of the cylinder and cylinder head are thus cooled by the exhaust steam, and when live steam is again admitted to the cylinder

* In the latest type of this engine, the lay shaft is omitted, the admission valves are operated by an eccentric controlled by a shaft governor, and the exhaust valves by a fixed eccentric.

† This type of engine is also called *uniflow.*

it is cooled by coming in contact with the cooler surfaces of the cylinder, and some of it is condensed.

To decrease this condensation of steam which is known as **initial condensation,** the unaflow engine is used. In this engine steam enters the cylinder at the ends and, after cut-off and expansion have taken place, is

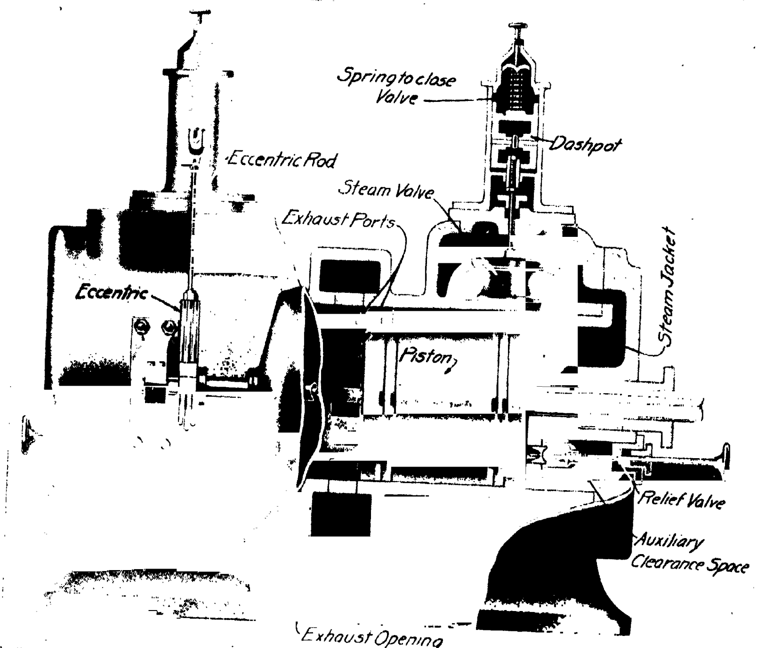


Fig. 295. — Unaflow Engine Cylinder.

exhausted through ports arranged around the center of the cylinder and uncovered by the piston near the end of the stroke. The steam thus flows in one direction through the cylinder, and the cylinder walls and heads are not cooled by the exhaust steam.

The cylinder of the unaflow engine, Fig. 295, is made slightly different from the cylinders already considered, in order to accommodate the valves, and the cylinder heads at each end are made with a passage for live steam to prevent radiation loss from the cylinder head. There are two steam poppet valves, operated as for the poppet-valve engine shown in Fig. 291. The exhaust valves are omitted, and their function is performed by the piston, which is made long for that purpose. The exhaust ports have a large area which permits the steam to escape rapidly, as the exhaust is only open for a short time. The exhaust passage is formed by a ring cast around the cylinder and connected to the exhaust pipe at the bottom.

The European unaflow engine is generally operated condensing; that is, the exhaust pressure is below the pressure of the atmosphere. Compression, beginning as soon as the piston covers the exhaust ports on its return stroke, takes place during 90 per cent of the stroke. When the engine is operating condensing, the final pressure will reach the initial steam-pipe pressure, but, when it is operated non-condensing, will rise above steam-pipe pressure with ordinary clearances. To prevent this rise in pressure when operating non-condensing, the following methods are employed:

1. Increasing the clearance by using clearance pockets in cylinder heads, controlled by hand-operated relief valves as illustrated in Fig. 295, or by concaving the ends of the piston.
2. Using automatic relief valves.
3. Using exhaust ports in the piston.
4. Using auxiliary exhaust ports to delay compression.

The additional clearance necessary varies with the steam pressure and for low steam pressures may be as high as 20 per cent.

The Skinner unaflow engine shown in Fig. 296 is a high speed engine which runs at speeds ranging from 250 to 275 r.p.m. When operating

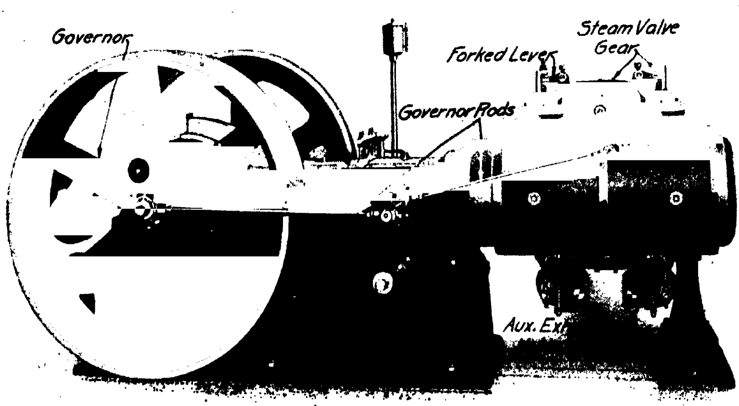


FIG. 296. — Skinner Unaflow Engine-Valve Gear Side.

condensing, an excessive pressure during compression is preventea by using auxiliary exhaust ports, located near each end of the bore of the cylinder, as shown in Fig. 297, where the piston and double-beat poppet valves are in such a position that *exhaust* is taking place at the crank end and *admission* at the head end.

The steam valve gear consists of two lifter bellcranks operated by a rocker arm located in a case containing oil and moved by a rocker and reach rod connection to the inertia type governor. One end of each lifter bell-

crank supports a hardened steel roller, which at all times bears against the cam on the rocker bar, and the other end makes contact with a set screw

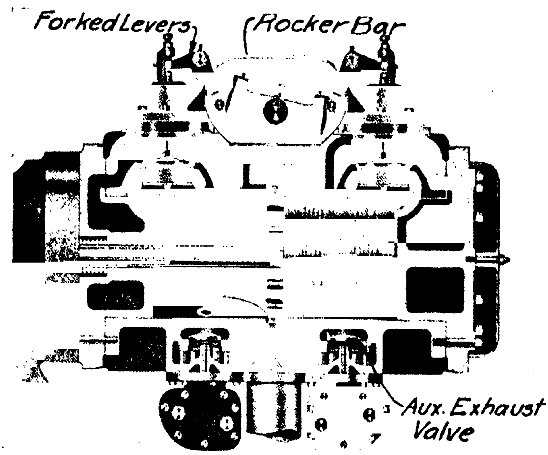


FIG. 297. — Cylinder of Skinner Unaflow Engine.

at the upper part of the valve stem for all positions before that of cut-off, after which there is a small clearance in order to permit the valve to close tightly. The inlet valves are thus opened by the cams on the rocker bar and are closed by a spring located in the valve bonnet and surrounding the valve stem, which is grooved and made a close fit in the sleeve, in which it slides, in order to prevent leakage of steam at this point. The position of the rocker is controlled by the governor, which changes the amount of movement and position of the cams, thereby changing the point of cut-off to meet the load requirements.

The **auxiliary exhaust valves** are designed to open and close under no difference in pressure, and are operated by a cam driven by a fixed eccentric located on the opposite side of the engine from the steam valve gear. These valves are of the poppet type and are so arranged that the engine may be

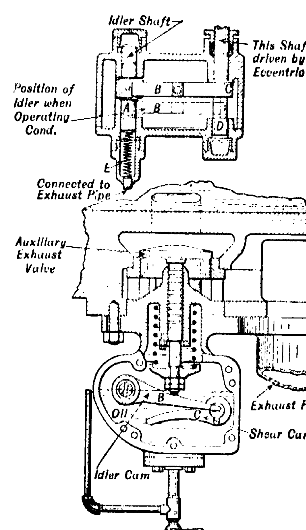


FIG. 298. — Automatic Device to operate Auxiliary Exhaust Valve of Skinner Unaflow Engine.

operated either condensing or non-condensing, without loss of efficiency, by means of an automatic disengaging device.

When operating condensing, the auxiliary exhaust valves are not used; when operating non-condensing, the auxiliary exhaust valves are automatically opened by the device shown in Fig. 298. An idler is located on a shaft which is free to move axially under the action of a spring located in a pocket connected to the exhaust pipe. A shear cam, operated by the engine valve gear, through a connection to the shear cam shaft, raises the exhaust valve when in the position shown. The spring around the valve stem has just enough tension to insure quick closing at high speeds.

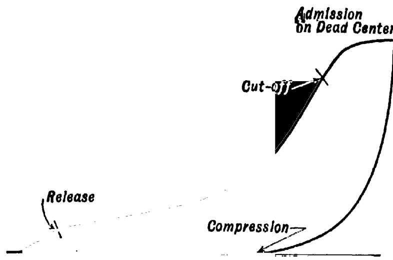


FIG. 299. — Indicator Diagram from Skinner Unaflow Engine.

Under vacuum, the tension in the spring on the idler shaft is overcome, the idler is moved out of register with the cam driven by the valve gear, and the exhaust valve remains closed. Should the vacuum be lost, the

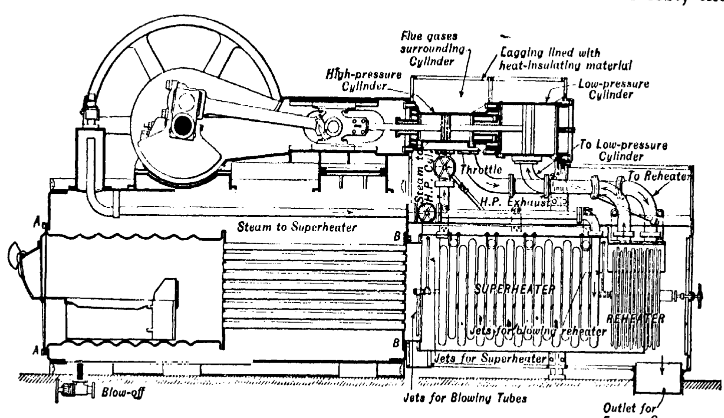


FIG. 300. — Buckeye Locomobile — Sectional View.

spring moves the idler between the valve stem and cam, the auxiliary exhaust begins to function, and the engine automatically operates non-condensing. A typical indicator diagram is shown in Fig. 299.

The unaflow engine manufactured by the Nordberg Mfg. Co. uses a long-range high-speed Corliss valve gear for the admission valves, and additional clearance is obtained by using clearance pockets.

379. Locomobile Engine. — The locomobile, Fig. 300, which is constructed in sizes up to 800 horsepower, is essentially a complete, efficient power plant. The engine is compound, that is, has two cylinders, the steam passing from one into the other and performing work in both, and is mounted upon the shell of an internally-fired boiler. The cylinders of the engine are located in the smoke box. Steam passes from the boiler,

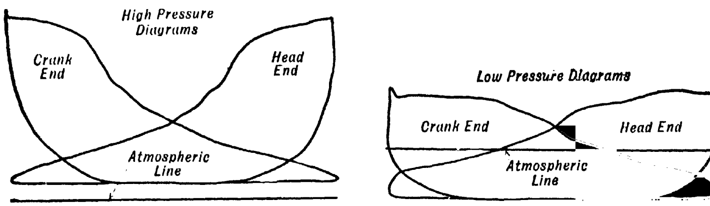


FIG. 301. — Indicator Diagrams from Buckeye Locomobile.

through the superheater, and into the high-pressure cylinder. From the high-pressure cylinder the steam passes through a **reheater,** directly to the low-pressure cylinder, thence to the condenser. The feed and condenser pumps are run by the main engine. Gases from the furnace pass back around the superheater and receiver, and then rise around the cylinders to the smoke connections at the top. The stuffing boxes for the piston and valves are of special construction to withstand the high temperatures to which they are exposed.

The governor is a centrifugal inertia governor, Art. 349, page 334, and controls the steam piston valves on the high-pressure cylinder only. Typical diagrams from the locomobile engine are shown in Fig. 301.

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REVIEW QUESTIONS

1. What advantage is obtained by using four valves in place of one, to control the distribution of steam?

2. Name the various types of Corliss valve gears.

3. Explain the method of operation of the standard Corliss valve gear.

4. To about what proportion of the stroke is the cut-off limited on a single eccentric Corliss engine?

5. Mention two methods used to increase the proportion of the stroke at which cut-off can occur on a single eccentric Corliss engine.

6. Explain how the Corliss long-range valve gear differs from the standard Corliss valve gear.

7. What is meant by a poppet-valve gear?

8. Describe the Lentz poppet-valve gear.

9. What is the advantage of a unaflow engine over a double-flow engine?

10. Describe the method employed to change the operation of the Skinner unaflow engine from condensing to non-condensing.

CHAPTER XIX

STEAM ENGINE INDICATOR AND ITS APPLICATIONS
STEAM ENGINE EFFICIENCIES AND LOSSES

380. Foreword. — A steam engine **indicator** is an instrument which records graphically the variation in the pressure of the steam, occurring in the cylinder of an engine during the stroke of the piston. This graphical record, when made for one revolution of the engine, is a closed curve, called an indicator diagram. It is formed by (1) a horizontal movement of the paper in exact correspondence with the movement of the piston, and (2) a vertical movement of the pencil in exact ratio to the pressure exerted in the cylinder of the engine. *The length of the diagram, therefore, represents the length of the stroke to a reduced scale, and its height at any point represents the pressure on the piston at the corresponding point in the stroke.* The indicator diagram may be used for the following purposes:

1. To furnish data for calculating the power developed by an engine.
2. To supply information regarding the accuracy of the setting of the valves.
3. To estimate the theoretical amount of steam used by the engine.

381. Description of the Steam Indicator. — There are many makes of indicators which differ mainly in details of construction, the essential operating parts being nearly the same in all. Two typical makes will be described, to give information covering the most common types. Steam engine indicators may be classified as:

1. Inside spring.
2. Outside spring.
3. Continuous.

The essential parts of the steam indicator are as follows:

1. A piston arranged to move in a cylinder and upon which the steam pressure acts.
2. A pencil motion, made of a number of small links, arranged to move the outer end of a pencil arm in a direction parallel to the piston, and to amplify its movements.
3. A piston rod connecting the piston and the pencil motion.
4. A spring, interposed between the piston and pencil motion, to measure the pressure in pounds per square inch.
5. A drum, which can be rotated about its vertical axis and which carries the paper upon which the graphical record is made.

The indicator is ordinarily attached to a short length of pipe leading into the clearance space of the cylinder. Steam from the cylinder enters below the piston, compresses the spring and moves the pencil arm, which stands at a height proportional to the steam pressure in the engine cylinder. The drum, upon which the indicator card* is held, is rotated by a suitable con-

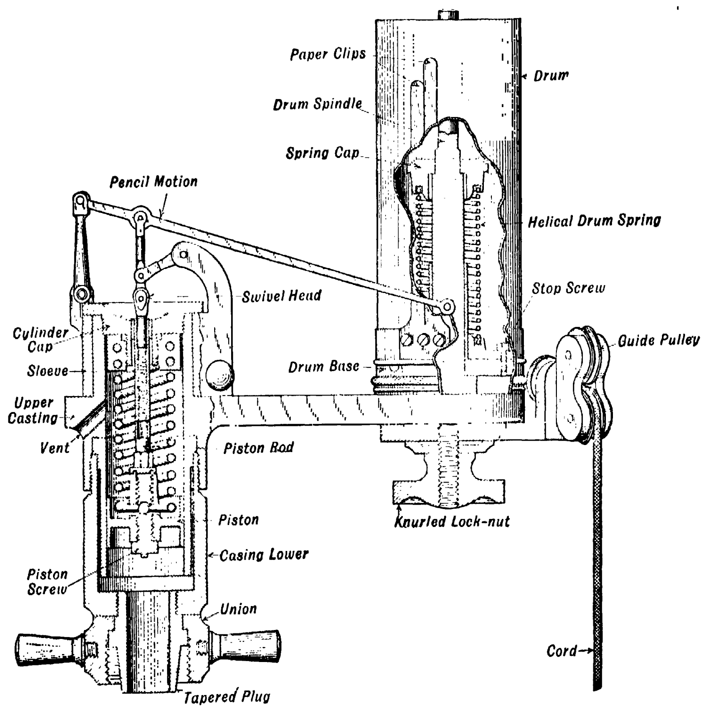


FIG. 302. — Sectional View of Crosby Inside Spring Steam Indicator.

nection to the crosshead. As the rotation occurs, the pencil is pressed against the indicator card and a line is drawn, which is produced by a combination of these two movements and which forms the diagram.

382. Crosby Inside Spring Indicator. — A sectional view of this instrument is shown in Fig. 302. The cylinder in which the piston slides is made of an alloy suited to the varying temperatures to which it is subjected. It is held between an upper and a lower casting, from which it is separated by a space, forming a steam jacket for the cylinder. The upper casting carries an arm, to the outer end of which the drum is attached, and its lower end is threaded inside to screw on the top part of the bottom casting. A

* The term *card* is here used to denote the paper upon which the diagram is drawn.

series of vent holes in this casting serve to maintain atmospheric pressure above the piston. A cap screws into the upper part of this casting and holds the sleeve carrying the pencil motion in place; in the center of the cap is a hole fitted with a hardened-steel bushing which guides the upper end of the piston rod. The lower casting carries a tapered sleeve and nut, at its lower end, by which the indicator is attached to the indicator cock.

The piston has an area of $\frac{1}{2}$ square inch, and is a hardened cylindrical tool-steel shell having a transverse web near its center, with a hub projecting above and below the web. The part of the hub above the web is threaded inside to receive the lower end of the hollow steel piston rod, and has a longitudinal slot into which fits a cross-wire on the lower end of the spring. The part of the hub below the web is threaded for a hexagonal-headed screw, having a concave bearing in its upper end. The lower end of the piston rod is threaded to screw into the hub of the piston, and above the threaded part is a shoulder having a circular channel, on its under side, which fits over a machined portion of the hub and prevents it from spreading. The upper end of the piston rod is threaded inside and screws on a threaded rod attached to the pencil motion. This permits changing the position of the pencil point by turning the piston rod with the attached spring. The fundamental principle of the pencil motion is that of a **pantograph parallel motion.**

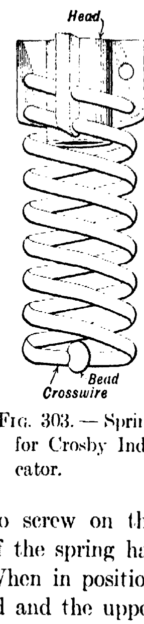


FIG. 303. — Spring for Crosby Indicator.

The spring, Fig. 303, is made of a single piece of spring-steel wire wound from the middle into a double coil, and fastened to a metal head threaded inside to screw on the inner threaded portion of the cap. The lower end of the spring has a cross-wire, midway on which is staked a small **bead.** When in position this bead is held between the lower end of the piston rod and the upper end of the piston screw. By this connection a ball and socket joint is formed, which prevents binding of the piston.

Indicator springs are made of various sizes of wire, to give a satisfactory height of diagram when used with different steam pressures. A number, known as the scale of the spring, is stamped on one of the flanges of the spring. *This number represents the pressure in pounds per square inch which would compress or elongate the spring sufficiently to move the pencil point one inch.* The most common scales of springs used in steam indicators are: 4, 8, 12, 16, 20, 30, 40, 50, 60, 80, 100, 120, 150, and 180.

The scale of the spring should be such that the diagram will be about $1\frac{3}{4}$ inches high.

The **drum** is a hollow, thin metal cylinder, closed at one end. Its open

end fits tightly over a hub which turns on a bearing formed by the **drum spindle.** The lower end of the spindle is screwed into the supporting arm, with its upper end projecting through a hole in the top of the drum. A guide pulley bracket, held in place by a knurled lock-nut, fits over the screw part of the spindle below the supporting arm. The drum is rotated against the action of the drum spring by a cord wound around the hub of the drum. By changing the position of a small screw in the hub, the instrument can be prepared for use on either right- or left-hand engines.

The drum spring is helical with its lower end attached to the hub, and the upper end to a cap with a square hole which fits over a squared shoulder on the drum spindle. The tension of the drum spring can be changed, as required for the speed, by lifting the cap and turning in the proper direction. The higher the speed the greater the tension required.

A small handle, adjustable by a screw against a stop, is attached to the sleeve carrying the pencil motion, and the pressure of the pencil against the paper on the drum can thus be regulated. *It should be sufficient to give a clear line with minimum friction on the paper.*

Method of attaching spring. — To place a spring in the instrument the cap is unscrewed, and the pencil motion, sleeve, piston, and piston rod are lifted from the cylinder. The piston rod is unscrewed from the piston and pencil motion, and the piston-screw backed part way out. The piston rod is inserted into a hollow wrench provided for this purpose, and the rod and wrench inserted into the spring, until the bead on the cross-wire rests on the concave end of the rod. The piston is then screwed on the rod as far as it will go, or until the upper end of the hub is brought against the bottom of the channel. The piston-screw is tightened and the piston rod screwed on the threaded part of the swivel head of the pencil motion, until the head of the spring is firmly fastened to the cap. The cap is then released and the turning continued until the top of the piston rod is flush with the shoulder at the swivel head. *In this position the atmospheric line should be at the proper height. If it is desired to change its position, the piston and cap are turned clockwise to lower, and counter clockwise to raise the atmospheric line. One turn changes the position of the pencil point $\frac{1}{8}$ inch.* The piston and its connecting parts are now placed in the cylinder, and the cap screwed down. The cap should always be examined before steam is turned on, to make sure it is properly screwed in position.

383. Thompson Inside-spring Indicator. — This instrument, Fig. 304, differs somewhat from the Crosby indicator, and is of more rugged construction throughout. The pencil motion is a **Watt parallel motion,** and the drum spring is a **flat spiral,** the tension of which is controlled by an adjusting disk and a lock-nut with a left-hand thread. The indicator spring is made of a single helical coil of wire fastened at each end to **inside-threaded ferrules;** one end of the spring screws on the piston and the other

on the cap. The piston rod is made of several parts, Fig. 305; its lower part is solid and is screwed firmly into the hub of the piston, which is made

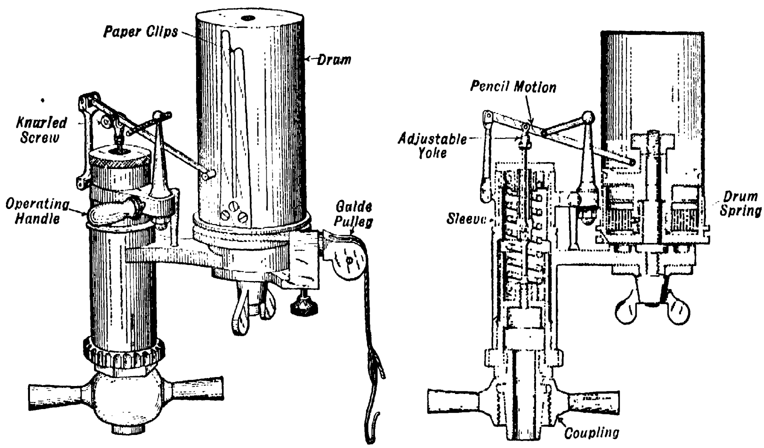


FIG. 304. — Outside and Sectional Views of Thompson Inside Spring Steam Indicator.

of brass and grooved to minimize leakage of steam. The upper part is made hollow and is threaded to screw over the threaded upper portion of

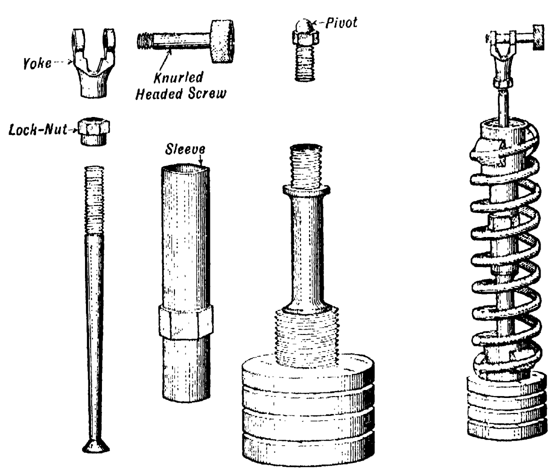


FIG. 305. — Indicator Piston and Spring for Thompson Indicator.

the lower part. It slides in a hole in the cap, and acts as a crosshead to guide the piston. A small rod is attached at its upper end to the pencil

motion by an adjustable yoke, and at its lower end is shaped to fit between the upper and lower parts of the piston rod to form a flexible joint. The atmospheric line is adjusted by changing the position of the yoke on its rod. The spring is easily placed in this instrument by unscrewing the cap and lifting the attached parts from the cylinder. After the small pin, with knurled end, which fastens the yoke to the pencil arm, has been unscrewed, the spring may be screwed firmly to the piston and then to the cap. The yoke is again attached to the pencil arm by the pin, and after the cap has been screwed in position, the operation is complete.

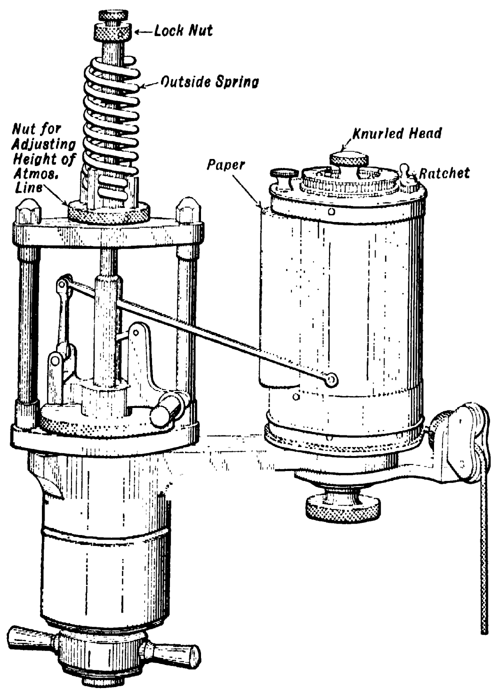


FIG. 306. — Crosby Outside Spring Indicator for Taking Continuous Diagrams.

This instrument is made for either right- or left-hand engines.

384. Outside-spring Indicator. — The new Crosby outside-spring indicator, Fig. 306, is illustrative of this type of instrument. The spring is outside the cylinder and is always at room temperature. The piston is made narrow and is, in form, the central portion of a sphere. It has an area of one square inch and the indicator springs are consequently made of larger-sized wire than those for inside spring indicators. The spring is easily attached to an extension of the piston rod which extends above the spring supporting frame, by unscrewing the adjusting screw and lock-washer and screwing the threaded portion of the spring to the spring support, with the bead on the cross-wire in the slotted upper end of piston rod extension. With this done, the small nut at the top is screwed against the bead and locked. The atmospheric line is changed by changing the position of the **lock-washer,** against which the lower end of the spring rests when in position.

385. Continuous-diagram Indicator. — To obtain continuous records over an extended period of time, an indicator must be used that will permit the taking of diagrams in rapid succession. Fig. 306 shows such a device attached to the drum of a Crosby indicator. A roll of paper 12 feet long and 2 inches wide is located on a spindle within an opening in the shell of the drum. The paper passes from this roll around the drum and inward

Fig. 307. — Continuous Indicator Diagram.

to a central cylinder, concentric with the shell of the drum, to which it is attached. Upon the top of the drum is a **ratchet wheel** which automatically unwinds a small length of paper from the roll and winds it on the inner cylinder, thus giving a series of diagrams which overlap each other slightly, as shown in Fig. 307. A **knurled head,** loosely attached to the drum spindle at the top, can be adjusted to move the paper around the drum by different amounts as desired, thus providing a method of controlling the distance between diagrams.

386. Indicator Piping and Cocks. — The indicator is connected to the cylinder by an indicator cock, Fig. 308*a*, and short ½-inch pipe. *The piping connections should be as short and direct as possible, with few bends, to reduce condensation and loss of pressure to a minimum.* The **straightway indicator cock** consists of a casting threaded at both ends and having a

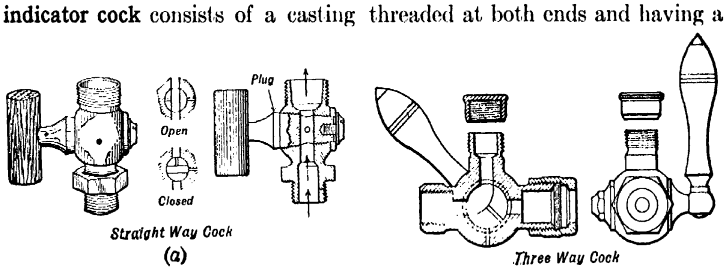


Fig. 308. — Indicator Cocks.

circular passage from end to end. At right angles to this passage is a **conical plug** having a wooden handle The upper end of the passage in the casting is tapered to form a tight joint with the tapered portion of the indicator coupling. With the handle of the plug parallel to the pipe, a straight passage is made to the cylinder. When the handle is at right angles to the pipe, the passage to the engine is closed, and the indicator is connected to the atmosphere by a small hole at the side of the casting.

A **three-way cock,** Fig. 308*b*, is sometimes used with a single indicator for both ends of the cylinder; it is better, however, to have a separate

indicator for each end of the cylinder. With the handle of the three-way cock vertical and toward the indicator, the passage between the indicator and cylinder is closed. When swung to the right, the left connection to the cylinder is open. In general, the opening is to the end opposite that toward which the handle is swung in opening.

387. Reducing Motions. — *A reducing motion is used to reproduce the motion of the crosshead on a smaller scale.* The drum of the indicator is attached to the reducing motion by a cord, which should be as short as possible, and thus it is made to move with the crosshead. *The number of inches of the travel of the piston represented by one inch in length on the diagram is called the* **ratio of reduction.** To be satisfactory, the reducing motion should be proportioned to give a length of diagram ranging from $2\frac{1}{2}$ to 4 inches. The reducing motion should be accurate, all its joints should be free of looseness, and its parts should be light and rigid.

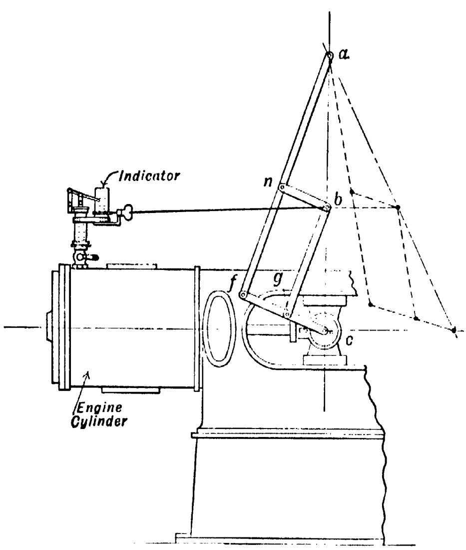


FIG. 309. — Pantograph Reducing Motion.

The **pantograph reducing motion,** Fig. 309, is well suited for slow moving engines. The indicator cord must be attached to the device at a point lying on the broken line *ac*, connecting the fixed point and the crosshead connection. *The ratio of reduction is then af divided by nf.* The four-bar linkage, *fnbg*, should be an exact parallelogram, and the chord to the drum should be parallel to the center line of the engine when leaving the linkage, or the reduction will not be in the above ratio.

The **reducing wheel,** Fig. 310, is portable and easily manipulated. The indicator is attached directly to the wheel casting, which in turn is attached to the indicator cock. The cord from the indicator passes around a small pulley, driven from the crosshead by a string wound around the larger, or driving, pulley. The movement of the driving pulley is communicated to the drum-cord pulley through a set of spur gears. *The ratio of reduction is the number of teeth in the driven gear divided by the number of teeth in the driving gear multiplied by the diameter of the driving pulley divided by the diameter of*

the driven pulley. This ratio may be changed by changing the pulley diameters or the size of the respective gears. A spring located in the spring case keeps the driving cord taut.

388. Accuracy of Indicator Diagrams. — It is possible to obtain accurate diagrams only when the indicator satisfies the following conditions:

1. Equal changes in pressure should be accompanied by equal movements of the pencil point. This requires an accurate pencil motion and spring.

2. Equal distances on the diagram should correspond to equal distances traveled by the piston. This requires a true cylindrical drum turning on its true geometric axis, a cord that does not stretch, and an accurate reducing motion.

389. Method of Taking Diagrams. — A satisfactory method of taking indicator diagrams is as follows:

1. The indicator cocks are opened and the steam is permitted to blow out any dirt, after which the indicator is attached and the length of the indicator cord adjusted to prevent the indicator drum striking the drum stop-pin at either end of the stroke. The indicator cord should always leave the reducing motion parallel to the center line of the engine cylinder, and the hook on the cord should be so attached that it will not slip and yet will be easily loosened.

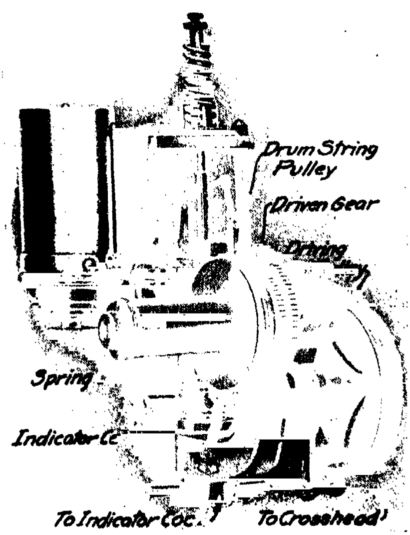


FIG. 310. — Crosby Reducing Wheel.

2. The tension of the drum is adjusted for the speed and the paper placed on the drum by folding one end over about ¼-inch, and hooking this bend over the longer of the paper clips; the paper is then looped around the drum and under the shorter clip, and with the forefinger and thumb the card is pulled down on the drum, at the same time it is kept tight against the drum by pulling outward on the ends held between the thumb and finger. The paper may have a prepared or a plain surface. The prepared paper requires a **brass stylus,** which will make a black line on the **zinc oxide** coating of the paper, while the plain paper requires only the lead of a pencil. The coated side of the prepared paper should be outward.

3. The pencil is adjusted against the paper to give a fine clear outline without undue friction on the paper.

4. The indicator cord is attached to reducing motion and the indicator cock partially opened to permit steam to blow through the cock into the atmosphere, thus removing water from the pipe connections to the indicator.

5. The indicator cock is next opened wide and the pencil point is pressed against the paper. If the pencil is held against the drum for too long a time, several diagrams will be obtained, in case the load is varying.

6. The indicator cock is now closed and the pencil pressed against the paper, to draw the atmospheric line.

390. Indicated Horsepower, i.hp. — *The indicated horsepower is the power developed in the steam cylinder, as determined from the indicator diagram.*

Horsepower, as previously defined, equals the product of a force in pounds multiplied by the velocity in feet per minute at which the force is moving, divided by 33,000 or, as given in Art. 54, page 68,

$$\text{Horsepower} = \frac{F \times V}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

The force acting on the piston of a steam engine changes from point to point. *Its mean value in pounds per square inch is obtained from the indicator diagram, by dividing the area of the diagram by its length and multiplying by the scale of the indicator spring.* This pressure is called the **mean effective pressure,** or **m.e.p.** The **total force** equals the mean effective pressure, P, in pounds per square inch multiplied by the net area of the piston, A, in square inches, against which the pressure acts. *It should be noted that the area of the crank end of the piston is less than that of the head end by the area of the piston rod.*

The velocity of the piston in feet per minute equals the length of the stroke in feet, L, multiplied by the number of working strokes per minute, N. Substituting these values of force and velocity in Equation (80), there results

$$\text{Indicated horsepower, i.hp.} = \frac{PLAN}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad (81)$$

For a double-acting engine, the total power equals the sum of the indicated horsepower for each end, or

$$\text{Total i.hp.} = \frac{P_h L A_h N}{33{,}000} + \frac{P_c L A_c N}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

in which P_h and A_h refer to the head end and P_c and A_c to the crank end. In these equations for indicated horsepower, $\frac{LA}{33{,}000}$ is constant for a given

engine and is known as the **engine constant.** When making a large number of calculations, the engine constant *should be computed for each end of the cylinder.* The indicated horsepower is then found by multiplying the engine constant by the mean effective pressure and the number of revolutions per minute.

It should be noted that the actual mean effective pressure is not the average pressure acting on one side of the piston. It is the average pressure acting on one side of the piston less that acting on the opposite side.

Example 37. — The following data were taken from a 12 in. by 24 in. Allis-Chalmers Corliss engine running at 102.9 r.p.m.: diameter of piston rod $2\frac{3}{16}$ in.; scale of indicator spring, 80 lb.; area of head-end diagram, 2.04 sq. in.; area crank-end diagram, 1.85 sq. in.; length of each diagram, 3.76 in.

Find: (*a*) Engine constant for each end, (*b*) total indicated horsepower.

Solution. — (*a*) Head-end constant .... $= \dfrac{L_h A_h}{33{,}000} = \dfrac{24 \times 113.10}{12 \times 33{,}000} = 0.00685$

Crank-end constant $= \dfrac{L_c A_c}{33{,}000} = \dfrac{24 \times (113.10 - 3.76)}{12 \times 33{,}000} = 0.00664$

(*b*) Head-end i.hp. .... $=$ Head-end constant $\times N \times P$

$= 0.00685 \times 102.9 \times 43.40 = 30.5$

Mean effective pressure $= \dfrac{\text{area of diagram} \times \text{scale of spring}}{\text{length of diagram}}$

$= \dfrac{2.04 \times 80}{3.76} = 43.40$

Crank-end i.hp. .... $= 0.00664 \times 102.9 \times 39.40 = 26.8$

Crank-end m.e.p. ...... $= \dfrac{1.85}{3.76} \times 80 = 39.40$

Total i.hp. $= 30.5 + 26.8 = 57.3.$

391. Method of Finding the Area of an Indicator Diagram with a Planimeter. — The area of an indicator diagram is found by means of a

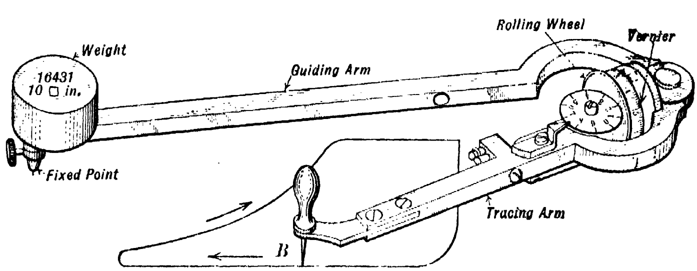


FIG. 311. — Fixed Arm Polar Planimeter.

planimeter, Fig. 311, which is an instrument used to obtain the areas of irregular figures. It has a **guiding arm** joined to a **tracing arm** by a hinged joint. The guiding arm has a point used as a pivot and held in a fixed position by a small weight. The tracing arm carries a **tracing point,**

which is moved over the outline of the figure to be measured. Fixed to the tracing arm is a small calibrated wheel, which revolves as the arm is moved. This wheel is divided into ten equal parts, each part representing one square inch. A complete revolution of the measuring wheel measures an area of 10 square inches. The area of the figure equals the difference in the readings on the recording wheel and vernier taken at the beginning and end of tracing around the figure.

The **vernier,** Fig. 312, is an auxiliary scale placed alongside the main scale to make it possible to secure an accurate reading of the hundredths place, which otherwise could only be estimated. The vernier scale is generally equal to the length of nine divisions on the main scale, and is divided into ten equal parts. Each vernier scale division is therefore one-tenth of a scale sub-division shorter than a sub-division of the main scale. To read, locate the scale mark before the index, and then look forward until a line of the vernier coincides with a line on the scale. Record the vernier reading as the last digit. The reading shown is **3.43.**

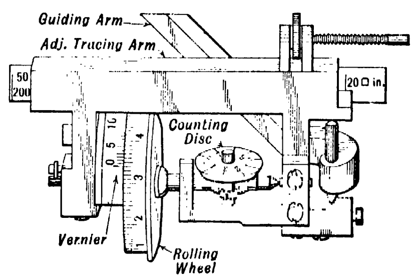


FIG. 312. — Rolling Wheel Showing Vernier.

The planimeter shown in Fig. 311 has a fixed tracing arm. Planimeters often have adjustable tracing arms which must be set to the proper scale before using. The area, as given by the planimeter, of a figure drawn to scale, must be corrected for the scale to which the figure is drawn, to obtain the actual area.

The indicator card, on which the diagram to be measured is drawn, should be fastened, by thumb tacks, to a piece of paper having a smooth unglazed surface for the wheel to roll upon; and the fixed point on the guiding arm should then be located so that the wheel will not come in contact with the edge of the card. The best location for the fixed point is above the card and with the arms making approximately 90 degrees with each other when the tracing point is midway between its extreme right and left positions on the figure.

A clean-cut mark, *B*, Fig. 311, is first made on the diagram. The tracing-point is placed on this mark and the reading on the wheel and vernier taken. The tracing point is then moved **clockwise** over the outline of the diagram to the starting point and a second reading taken. The difference in these two readings is the area in square inches. *Care should be taken to start and stop at the same point.*

The length of the indicator diagram is obtained by drawing fine lines

perpendicular to the atmospheric line from the extreme ends of the indicator diagram, and measuring the length between these lines on the atmospheric line. The area, as found above, divided by the length and multiplied by the scale of spring equals the *mean effective pressure.*

When a planimeter is not available, the mean effective pressure may be found by dividing the diagram into 10 equal parts, and then measuring the height of lines drawn at the center of each area enclosed by the diagram. *The sum of these heights, if all points on the expansion curve are above the back pressure line, divided by* 10 *and multiplied by the scale of spring will be the mean effective pressure* If there are points on the expansion line below the back pressure line, the corresponding heights must be subtracted instead of added.

392. Brake Horsepower. — *The brake horsepower is the power delivered by the crank shaft of the engine, as determined by a brake mounted on the flywheel or brake drum. The power is absorbed by the brake as friction and appears as heat.*

A typical **Prony brake,** shown in Fig. 313, consists of an arm, one end of which bears on a standard that rests on a scale. The opposite end is shaped

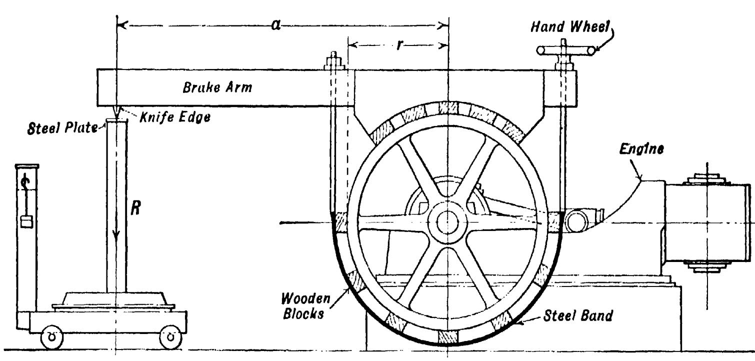


Fig. 313. — Prony Brake applied to Steam Engine.

to fit on the flywheel or brake drum, and supports an adjustable band, which passes around the pulley. The amount of friction, and consequently the load on the engine, is controlled by tightening or loosening the band.

The power absorbed by a brake is obtained by using the fundamental equation

$$Horsepower = \frac{force \times velocity\ per\ minute}{33{,}000}$$

Let r = radius of the brake drum measured in feet.

N = revolutions of engine shaft per minute.

F = resistance at circumference of wheel of radius r, lb.

R = gross load on scale, lb.
w = tare weight on scale, lb.
W = net load on scale, lb. = $R - w$.
a = length of brake arm in ft. = horizontal distance between center of brake-drum shaft and the point of support at scale.

The work absorbed in friction equals the force times the distance. Expressed as an equation,

$$\text{Work of friction} = F \times 2\pi r N \quad . \; . \; . \; . \; . \; . \; . \quad (83)$$

The force, F, is not known. The equivalent load, W, acting at the end of the brake arm of length a, may be used instead, since by the principle of moments $Fr = Wa$.

Placing these values in the fundamental horsepower equation, there results

$$\textit{Brake horsepower, b.hp.} = \frac{2\pi aWN}{33{,}000} \quad . \; . \; . \; . \quad (84)$$

The **brake tare** is the effective weight of the brake arm resting on the scale with the brake band loose. Its value is obtained by revolving the flywheel first forward and then backward, reading the load on the scale in each case. The speed in each direction should be the same. The weight on the scale when running forward is the weight of the arm acting on the scale plus friction; and when running backward, the weight of arm acting on the scale minus friction. Adding these weights eliminates the friction factor and gives twice the tare weight. For example; the weight when running forward is 38 pounds and backward 34 pounds; the tare weight then equals 38 plus 34 divided by 2, or 36 pounds. The **net scale load** is found by subtracting the tare weight from the gross scale reading.

Example 38. — An 8 in. by 12 in. steam engine is fitted with a Prony brake having an arm 54 inches long. During a test, the gross weight on the scale was 120 lb., the tare weight was 20 lb., and the revolutions per minute 221. Find the brake horsepower.

Solution. — Using Equation (84)

$$\text{B.hp.} = \frac{2\pi aWN}{33{,}000} = \frac{2 \times 3.14 \times 4.5 \times 100 \times 221}{33{,}000} = 18.9$$

$$a = \frac{54}{12} = 4.5 \text{ ft.}; \quad W = 120 - 20 = 100 \text{ lb.}; \quad N = 221$$

Friction horsepower. — The friction horsepower represents the power lost in friction and is found by subtracting the brake horsepower from the indicated horsepower.

393. Mechanical Efficiency. — This efficiency is the ratio of the power output of the engine, as measured by the brake, to the power developed by the steam in the cylinder of the engine, as obtained from the indicator diagrams. Expressed as an equation,

$$\textit{Mechanical efficiency} = \frac{b.hp.}{i.hp.} \quad . \; . \; . \; . \; . \; . \; . \quad (85)$$

Example 39. — Using the data from Example 38, in which the indicated horsepower was 21.7, find the mechanical efficency of the engine.

$$\text{Mechanical efficiency} = \frac{\text{b.hp.}}{\text{i.hp.}} = \frac{18.9}{21.7} = 0.87 \text{ or } 87 \text{ per cent.}$$

394. Weight of Steam used, as Shown by the Indicator Diagram. — When the diagram is used to calculate the theoretical amount of steam used, the results obtained are only approximate, since water may be mixed

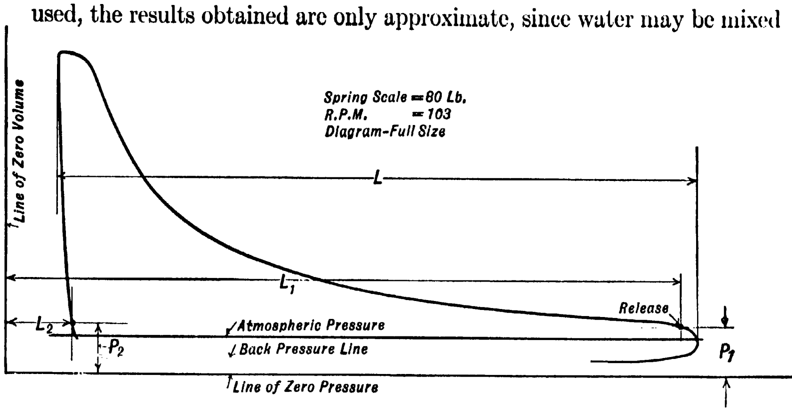


FIG. 314. — Indicator Diagram marked for Calculation of Diagram Water Rate.

with the steam at entrance, the steam condenses during admission, and steam leaks past the valves. This calculation is, however, customarily included in the calculated results of an engine test, and the method of making it should be understood, or incorrect conclusions may be drawn therefrom.

Reference is made to the indicator diagram shown in Fig. 314. Lines representing **zero volume** and **zero pressure** are drawn upon the indicator card and are laid off to the proper scale. The zero-volume line is located by multiplying the length L of the diagram by the per cent clearance and laying it off as indicated, and the zero-pressure line is located by measuring to the proper scale, below the atmospheric line, a distance corresponding to the pressure of the atmosphere. At the point of cut-off or at release, there is a certain weight of steam represented by the volume of the cylinder up to that point, including the clearance volume. At the point of compression there is a weight of steam represented by the volume enclosed between the piston and cylinder head. This weight is known as **cushion steam,** and its amount is found by multiplying the volume of the cylinder at the point of compression by the density of the steam corresponding to the absolute pressure at that point. The difference between the weight of steam at cut-off or release and at compression is the **cylinder feed.** It represents the weight of steam supplied per stroke.

To compute the **cylinder feed** as found from the diagram, it is necessary to choose the points of compression and release. The point of release is used instead of the point of cut-off since at that point the amount of re-evaporation, on expansion, is included in the volume, and the net result is nearer to the actual steam used than it would be if the volume and weight were taken at cut-off.

Let L = length of stroke, ft.
L_1 = length of the cylinder up to the point of release, including the distance represented by the clearance volume, ft.
L_2 = length of the cylinder up to the point of compression, including the distance represented by the clearance volume, ft.
N = revolutions per minute.
A = net area of piston, sq. in.

The volume of steam present at release $= L_1A \div 144$: and at compression $= L_2A \div 144$. The weight of the steam at release equals the volume present at release multiplied by the density of the steam, W_1, at the absolute pressure at release $= \dfrac{L_1AW_1}{144}$.

Similarly, the weight of steam at compression $= \dfrac{L_2AW_2}{144}$. The net weight of steam supplied per revolution $= \dfrac{L_1AW_1}{144} - \dfrac{L_2AW_2}{144} = \dfrac{A}{144}(L_1W_1 - L_2W_2)$.

The weight of steam for N revolutions per minute $= \dfrac{AN}{144}(L_1W_1 - L_2W_2)$.

The weight of steam used per indicated horsepower per hour is, therefore,

$$\frac{\dfrac{60\,AN\,(L_1W_1 - L_2W_2)}{144}}{\dfrac{PLAN}{33{,}000}} = \frac{13{,}750}{P}\left(\frac{(L_1W_1 - L_2W_2)}{L}\right) \quad . \ . \quad (86)$$

The values corresponding to L, L_1 and L_2 may be taken from the indicator diagram, since lengths on the diagram are proportional to the length of stroke.

Example 40. — The following data were taken from a 12 in. by 24 in. Corliss engine running at 102.9 r.p.m.; clearance of head-end, 7.89 per cent; clearance of crank-end 7.40 per cent.

Head-end diagram data: L_1 = 4.03 in., L_2 = 0.38 in., P_1 = 22.4 lb. per sq. in. abs., P_2 = 28 lb. per sq. in. abs., L = 3.87 in., W_1 = 0.055 lb. per cu. ft., W_2 = 0.068 lb. per cu. ft., m.e.p. = 42.8 lb. per sq. in. Find the theoretical weight of steam per indicated horsepower per hour, from the diagram.

Solution. — Substituting in Equation (86)

$$\text{Weight of steam per i.hp. per hr.} = \frac{13750}{\text{m.e.p.}} \frac{(W_1L_1 - W_2L_2)}{L}$$

$$= \frac{13750}{42.8}\left(\frac{0.055 \times 4.03 - 0.068 \times 0.38}{3.87}\right) = 16.20 \text{ lb.}$$

For the crank end, by a similar calculation, the weight of steam per i.hp. = 13.30 lb.

$$\text{Weight of steam per i.hp. per hour for whole engine} = \frac{16.20 + 13.30}{2} = 14.75 \text{ lb.}$$

A method of estimating the weight of steam used by a steam engine, by means of the indicator diagram, is reported in the TRANS. AM. SOC.

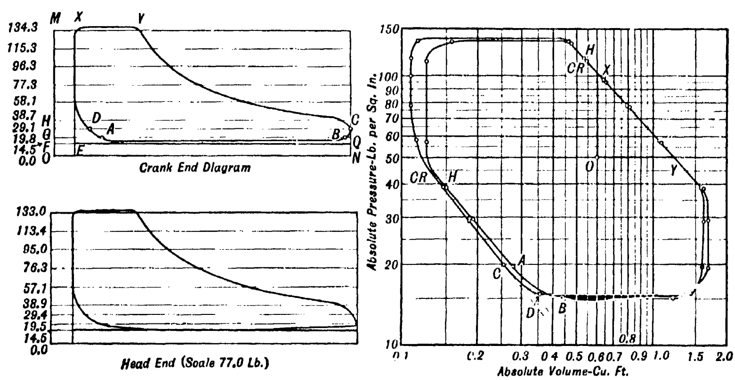
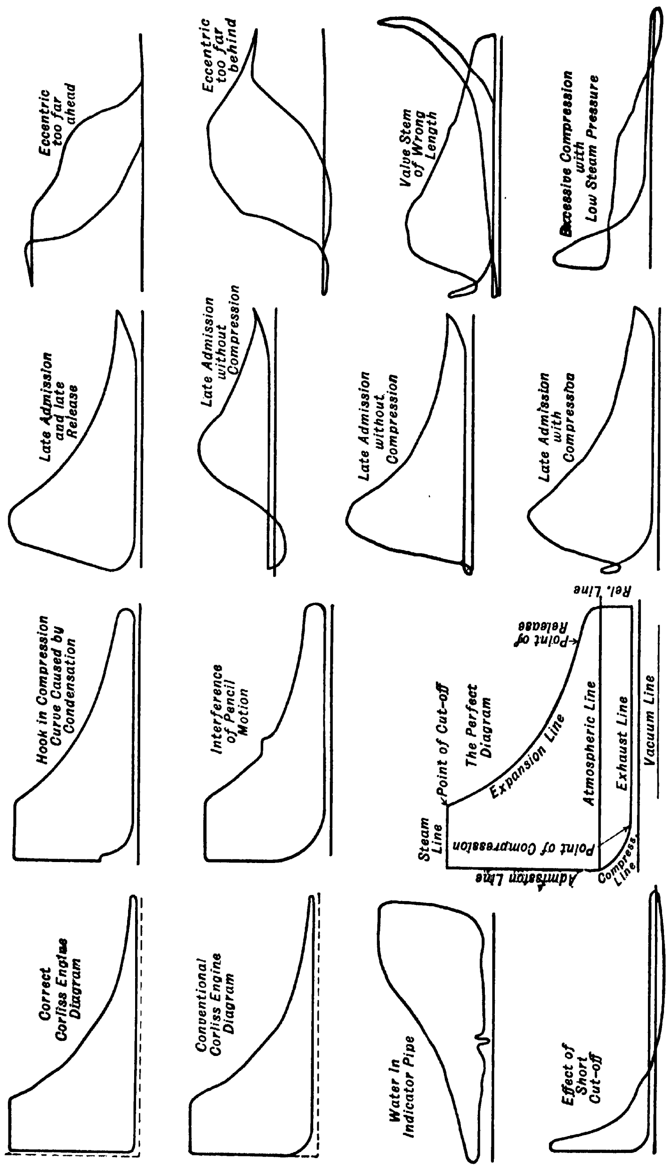


FIG. 315.—Original Indicator Diagram and Logarithmic Diagrams Plotted from them.

M. E., April, 1912, and is known as **Clayton's analysis.** It is claimed that this analysis accounts for the effect of initial condensation, and that results within 4 per cent of the weight of steam used, as determined by a test, can be obtained. The indicator diagrams are transferred to logarithmic cross-section paper, as shown for one diagram in Fig. 315. The expansion and compression lines then become straight lines, and the value of n, in the equation $PV^n = C$, can be obtained from the slope of the lines. Using the value of n thus obtained for the expansion curve, and a curve showing the relation between experimentally determined values of the quality of steam at cut-off, x_c, and values of n for expansion, the actual weight of steam used per revolution may be approximated for engines with the same type of cylinder as was used in determining the x_c and n curve.

395. Valve Setting with an Indicator. — In setting valves by means of an indicator, the diagrams taken from both ends of the cylinder are made to show a proper distribution of steam and to be as nearly alike as possible, except for a slightly later cut-off on the crank end to allow for the reduction of piston area by the piston rod. For a slide-valve engine, this is done by



(Taken from Power.)

FIG. 316. — Corliss Indicator Diagrams showing Faulty Valve Settings.

adjusting the length of valve stem and changing the position of the eccentric until the diagrams appear satisfactory.

Typical indicator diagrams showing faulty valve settings are shown in Fig. 316. A careful study should be made of these diagrams as a guide to proper valve setting by the indicator method.

It should be remembered that the information furnished by the indicator diagram, regarding the setting of the valves, is purely a matter of inference.

It is often difficult to determine the point at which an event on the diagram occurs, because the curves run into each other gradually, without clearly defining the point of separation of the curves. The point can be easily located by producing both curves along their regular trend until the curves meet; the point of tangency is the desired point.

396. Theoretical Indicator Diagram. — The theoretical indicator diagram is shown in Fig. 317, by dotted lines, for both condensing and non-condensing operation. These diagrams differ from the actual diagrams, which are shown by solid heavy lines. An engine capable of giving the theoretical diagram would operate without clearance or compression; the valves would open and close instantaneously; the steam would enter the cylinder without drop in pressure during admission, and at the pressure in the boiler; the exhaust pressure would be that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine; and the expansion would be according to the equation PV = a constant. This equation is used because it is easier to construct than the adiabatic curve, to which the actual expansion curve, doubtless, more nearly coincides.

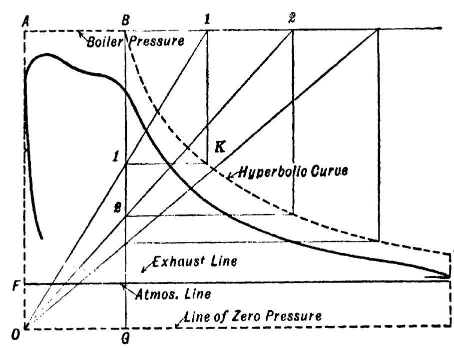


FIG. 317. — Diagram Factor Non-Condensing Engine.

The expansion curve, or **rectangular hyperbola,** may be constructed by drawing BG perpendicular to the atmospheric line through the point of cut-off, and also a horizontal line through the same point. From O, the point of intersection of the line of zero pressure and zero volume, diagonal lines are drawn cutting each of these lines, as at 1-1 and 2-2, and from these points of intersection are drawn horizontal and vertical lines. The point of intersection, K, locates one point on the curve. By drawing a number of diagonal lines, a sufficient number of points may be located to draw the curve.

397. Theoretical Mean Effective Pressure. — In the discussion of the actual steam indicator diagram, it was shown that the mean ordinate equaled the area divided by the length of the diagram. Applying the same reasoning to Fig. 317, there results

$$\text{Mean ordinate} = \frac{\text{Area } A\ B\ C\ E\ O - F\ D\ E\ O}{\text{Length } O\ E} \quad . \quad . \quad . \quad (87)$$

$$\text{Area } A\ B\ C\ E\ O = \text{Area } A\ B\ G\ O + \text{area } B\ C\ E\ G$$

$$= O\ A \times O\ G + P_1 V_1 \log_e \frac{V_2}{V_1}$$

Let $O\ A = P_1 =$ initial absolute pressure, lb. per sq. in.
$O\ G = V_1 =$ volume of cylinder at cut-off, cu. ft.
$C\ E = P_2 =$ absolute pressure at release, lb. per sq. in.
$O\ E = V_2 =$ volume of cylinder at release, cu. ft.
$D\ E = P_3 =$ absolute exhaust pressure, lb. per sq. in.

Substituting these symbols in Equation (87), there results

$$\text{Theoretical m.e.p.} = \frac{P_1 V_1 + P_1 V_1 \log_e \frac{V_2}{V_1} - P_3 V_2}{V_2}$$

Dividing this equation by V_1 and representing $\frac{V_2}{V_1}$ by r, the expression for the mean effective pressure becomes

$$\text{Theoretical m.e.p.} = \frac{P_1\,(1 + \log_e r)}{r} - P_3 \quad . \quad . \quad . \quad . \quad (88)$$

The ratio $\frac{V_2}{V_1} = r$ is known as the **ratio of expansion** and is the ratio of the volume of steam at the end of the stroke to the volume at cut-off. Since the area of the piston is constant, $\frac{V_1}{V_2} = \frac{L_1}{L_2}$, and the reciprocal of r is the fraction of the stroke completed at cut-off. With a cut-off occurring at one-quarter of the stroke, the number of expansions r would be $(1 \div \frac{1}{4})$, or 4.

A comparison of an actual and a theoretical diagram for the same conditions shows that the actual diagram is smaller than the theoretical, because of imperfections in the working of the actual engine. During admission there is a loss in pressure caused by **throttling** of the steam when passing through the restricted openings and also by **initial condensation.** The valves do not open and close instantaneously, and a loss results, as shown by the rounded corners at cut-off, release and compression. Release occurs before the piston has reached the end of the stroke, and compression starts before the piston has completed the exhaust stroke.

Since the actual diagram is smaller than the theoretical, the **probable m.e.p.** may be obtained from the **theoretical m.e.p.** by multiplying it by a **diagram factor,** which is equal to the actual m.e.p. divided by the theoretical m.e.p. It varies for different engines, as shown in Table 27.

Example 41. — An 8 in. by 24 in. Corliss engine runs at 110 r.p.m. Initial steam pressure 100 lb. per sq. in. gage; back pressure 14.7 lb. per sq. in. abs.; cut-off at one-quarter stroke. Find (*a*) the theoretical m.e.p., (*b*) the probable m.e.p. if the diagram factor is 85 per cent, and (*c*) the probable indicated horsepower.

Solution. — Using Equation (88),

$$(a)\ \text{Theoretical m.e.p.} = \frac{P_1(1 + \log_e r)}{r} - P_3$$

$$= \frac{114.7\,(1 + 1.38)}{4} - 14.7 = 53.6 \text{ lb. per sq. in.}$$

$$P_1 = 100 + 14.7 = 114.7 \text{ lb. per sq. in. abs.};\ r = 4$$
$$\log_e r = 2.3 \log_{10} r = 2.3 \log_{10} 4 = 2.3 \times 0.6020 = 1.38$$
$$P_3 = 14.7 \text{ lb. per sq. in. abs.}$$

(*b*) Probable m.e.p. $= 53.6 \times$ diagram factor $= 53.6 \times 0.85 = 45.4$ lb. per sq. in.

$$(c)\ \text{Probable horsepower} = \frac{PLAN}{33{,}000} \times 2 = \frac{45.4 \times 2 \times 50.27 \times 110 \times 2}{33{,}000} = 30.4$$

$$L = 24 \text{ in.} = 2 \text{ ft.};\ N = 110;\ A = \tfrac{1}{4}\pi d^2 = \tfrac{1}{4} \times 3.14 \times 64 = 50.27 \text{ sq. in.}$$

TABLE 27. — DIAGRAM FACTORS FOR SIMPLE ENGINES*

Type of Engine	Diagram Factor Per Cent
Simple, slide valve	55 to 90
Simple, Corliss	85 to 90
Compound, slide valve	55 to 80
Compound, Corliss	75 to 85
Triple expansion	55 to 70

* From "Heat Power Engineering," Hirshfeld and Barnard.

398. Rating of Engines. — It is customary for manufacturers to rate steam engines on a basis of the indicated horsepower, when the cut-off occurs at one-quarter stroke and with a specified steam pressure at the throttle valve.

399. Methods of Stating Steam-engine Performance. — The performance of steam engines is customarily stated in the following ways:

1. Weight of steam used, pounds per hour or per i.hp.-hour.
2. Heat consumption, B.t.u. per i.hp.-hr. or per i.hp.-minute.
3. Thermal efficiency, per cent.
4. Rankine cycle ratio, per cent.
5. Mechanical efficiency, per cent.
6. Duty, for pumping engines (Consult Art. 504, page 518).

400. Weight of Steam Used. — When stated as the weight of steam per unit of power, this method of stating engine performance is known as the **water rate** of the engine. The water rate is of small value when comparing engines, because of variations in the initial steam pressure, quality of steam, and back pressure. For small variations of these quantities, it is

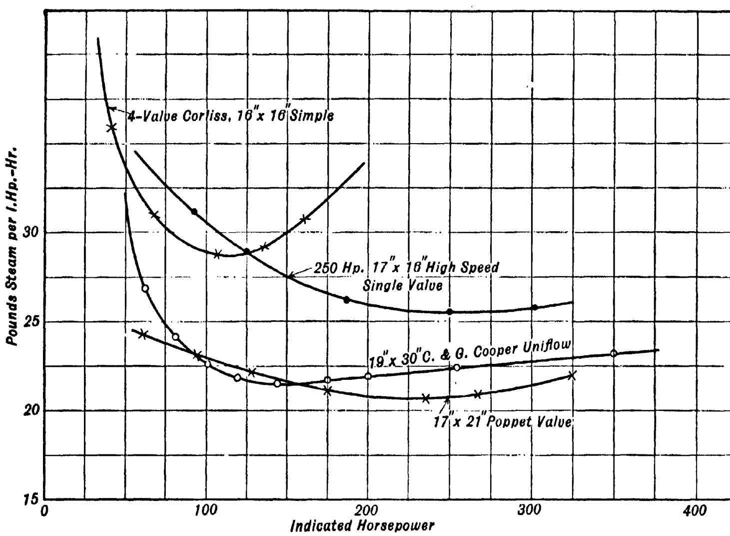


FIG. 318. — Typical Economy Curves — Non-Condensing Operation, Simple Engines.

customary to correct the water rate to conditions taken as standard. This should not be done when the variation in these quantities is over a few per cent. The variation in the water rate of a few typical engines, with the power output, is shown in Fig. 318. When the total weight of steam used per hour, by an engine having a throttle governor, is plotted against the indicated horsepower, it will show a straight line, known as **Willans line.**

401. Heat Consumption. — This affords the best method of comparing the performance of engines. The number of heat units consumed by the engine per hour is found by multiplying the weight of steam used in pounds per hour by the difference between the total heat in one pound of steam at the average absolute pressure and quality found in the steam pipe near the throttle valve, and the heat in one pound of water at the temperature of saturated steam at the average absolute pressure existing in the exhaust pipe near the engine.

Example 42. — The weight of steam used by an 8 in. by 18 in. Murray Corliss engine, during a test, was 780 lb. per hr.; steam pressure, 120 lb. per sq. in. gage; exhaust pressure, 0.10 lb. per sq. in. gage; barometer, 14.5 lb. per sq. in.; quality, 0.977.

Find (*a*) the heat consumption per hour, (*b*) the heat consumption per i.hp. when the i.hp. equaled 26.4.

Solution. — (*a*) Heat consumption = weight of steam per hour $\times$ $(x_1L_1 + h_1 - h_2)$
in which x_1 = quality of steam = 0.977

L_1 = latent heat of steam at initial abs. pressure = 872.25 B.t.u.
h_1 = heat of liquid at initial abs. pressure = 320.90 B.t.u.
h_2 = heat of liquid at abs. exhaust pressure = 180 B.t.u.
Absolute pressure at throttle = (120 + 14.5) = 134.5 lb. per sq. in.
Absolute pressure at exhaust = (0.10 + 14.5) = 14.6 lb. per sq. in.
Heat consumption = 780 (0.977 $\times$ 872.25 + 320.90 − 180) = 774,600 B.t.u.

(*b*) Heat consumption per i.hp. per hr. $= \frac{774{,}600}{26.4} = 29{,}300$ B.t.u.

402. Thermal Efficiency. — *The thermal efficiency of an engine is the ratio of the heat converted into useful work to the heat supplied, measured above the temperature corresponding to the pressure of the exhaust steam.* It is commonly based on the steam used per i.hp.-hour.

The heat converted into useful work equals the heat equivalent of the i.hp. per hour. Each horsepower hour is equivalent to $\frac{33000 \times 60}{775.4} =$ 2547 B.t.u. per hour.

The heat supplied equals $W\ (H_1 - h_2)$

in which W = pounds of steam as supplied, per i.hp.-hr.
H_1 = total heat above 32 deg. fahr. per pound of steam, at initial conditions prevailing before the throttle valve.
h_2 = heat of liquid above 32 deg. fahr. in one pound of water, at the temperature of saturated steam at exhaust pressure.

$$\text{Thermal efficiency} = \frac{2547}{W\ (H_1 - h_2)} \quad \text{(89)}$$

Example 43. — Using the data given under Example 42, page 400, find the thermal efficiency.

Solution. — By substituting proper values in Equation (89)

$$\text{Thermal efficiency} = \frac{2547}{29.5\ (1173.08 - 180)}$$
$$= \frac{2547}{29{,}300} = 0.087, \text{ or } 8.7 \text{ per cent}$$

403. Rankine-cycle Ratio, or Engine Efficiency. — This efficiency is sometimes called **potential efficiency** or **efficiency ratio.** *It shows the extent to which the performance of the actual engine, expressed in heat units, approaches the heat available, for an ideal engine working on a Rankine cycle.* This cycle is shown in Fig. 319, and is the accepted standard for comparing engine cycles.

To obtain an indicator diagram or cycle of this kind, the walls of the

cylinder and the piston would have to be non-conductors of heat; the expansion after cut-off would be adiabatic, that is, without receiving or giving up heat during the expansion and it would be continued until the pressure equaled the back pressure; the action of the valves would be instantaneous; the steam passages would be of sufficient area to prevent wire-drawing; the engine would operate without clearance; and all of the energy taken from the steam would be converted into useful work.

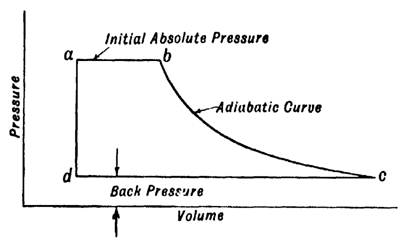


FIG. 319. — Diagram showing the Rankine Cycle.

The efficiency of the Rankine cycle may be expressed as an equation as follows:

$$\textit{Rankine efficiency} = \frac{\textit{Heat available}}{\textit{Heat supplied}} = \frac{H_1 - H_2}{H_1 - h_2} \quad . \ . \ . \quad (90)$$

in which H_1 = total heat above 32 deg. fahr. in one pound of steam at the absolute throttle pressure, B.t.u.

h_2 = heat of liquid at the temperature corresponding to the exhaust pressure, B.t.u.

H_2 = total heat above 32 deg. fahr. in one pound of steam after adiabatic expansion to the absolute exhaust pressure, B.t.u.

The value of H_2 equals $x_2L_2 + h_2$, and it may be found from the Steam Table, provided the quality at exhaust (x_2) is known. This quality may be found from the Mollier Chart, Fig. 76, page 100, or from the relation $\frac{x_1L_1}{T_1} + \theta_1 = \frac{x_2L_2}{T_2} + \theta_2$, since with an adiabatic expansion there is no heat change and the entropy at cut-off equals the entropy at release. In this equation, x = quality, L = latent heat, θ = entropy of liquid and T = absolute temperature. All these values, with the exception of x_2, may be found from the steam tables, as explained in Chapter IV, pages 88 to 100. Values of the efficiency of the Rankine cycle are given in Table 28 for different conditions of steam at inlet and exhaust. These values represent the maximum attainable thermal efficiency. For further study along this line, consult "Steam Power Plant Engineering" by GEBHARDT.

$$\begin{array}{c}\textit{The Rankine-cycle ratio}\\ \textit{referred to i.hp.}\end{array} = \frac{\textit{Thermal efficiency of the actual engine.}}{\textit{Thermal efficiency of the ideal engine.}}$$

$$= \frac{2547}{W\,(H_1 - h_2)} \div \frac{H_1 - H_2}{H_1 - h_2}$$

$$= \frac{2547}{W\ (H_1 - H_2)} \quad \ldots \ldots \ldots \ldots \quad (91)$$

in which the symbols are as defined in Arts. 402 and 403

TABLE 28. — RANKINE-CYCLE EFFICIENCIES AND THEORETICAL WATER RATES

Initial Pressure Lb. per Sq. In. Abs.	Dry and Saturated Steam				Superheated Steam		
	*Condensing		† Non-condensing		Superheat ° F.	Condensing	Non-Condensing
	Rankine Efficiency	Water-rate	Rankine Efficiency	Water-rate		Rankine Efficiency	Rankine Efficiency
50	29.48	8.98	8.98	28.51	.....	.....	.....
100	28.47	7.85	13.88	18.22	272.2	29.8	15.4
150	30.60	7.26	16.65	15.08	.....	.....	.....
200	31.88	6.94	18.60	13.44	218.1	32.9	19.7
250	32.93	6.70	20.05	12.42	.....	.....	.....
300	33.76	6.52	21.22	11.71	182.5	34.5	22.0
400	35.10	6.25	23.07	10.73	155.2	36.1	23.7
500	36.06	6.07	24.46	10.10	132.7	36.7	25.0
600	36.84	5.94	25.57	9.66	113.4	37.3	26.0

* Back pressure ½ pound per sq. in., abs. † Back pressure 14.7 lb. per sq. in., abs.

Example 44. — Data as given in Example 42, page 400. Find the Rankine-cycle ratio.

Solution. — Using Equation (91) and data as given,

$$\text{Rankine-cycle ratio} = \frac{2547}{W(H_1 - H_2)} = \frac{2547}{29.5\ (1173.08 - 1011.77)} = 0.533$$

$W = 29.5$ lb. per i.hp.-hr.

$H_1 = xL_1 + h_1 = 0.977 \times 872.25 + 320.9 = 1173.08$ B.t.u.

$H_2 = x_2L_2 + h_2 = 0.856 \times 971.7 + 180 = 1011.77$ B.t.u.

$$x_2 = \frac{\frac{x_1L_1}{T_1} + \theta_1 - \theta_2}{L_2 \div T_2} = \frac{0.977 \times 1.0774 + 0.5023 - 0.3120}{1.4469} = 0.856$$

Values of the Rankine-cycle ratio are given in Table 29 for a few typical steam engines, in order to give an idea of the value of this ratio for modern types of steam engines.

The data necessary to compute the various steam-engine efficiencies are obtained by making tests as explained in Chapter XXII.

404. Heat Losses in the Steam Engine. — It has been shown under engine efficiencies, that the actual steam engine has a lower efficiency than the ideal. This is caused by losses the most important of which are:

1. Cylinder condensation.
2. Leakage of steam.

3. Clearance volume.
4. Incomplete expansion.
5. Wire-drawing.
6. Friction of the engine parts.
7. Moisture in the entering steam.
8. Heat discharged in exhaust steam.
9. Radiation and other small losses.

Some of these losses are preventable, while others are inherent in the machine and cannot be prevented.

* TABLE 29. — VALUES OF RANKINE-CYCLE RATIO FOR TYPICAL STEAM ENGINES

Type of Engine	Ind. hp.	Size	Operating Conditions			Rankine Cycle Ratio	Water Rate
			Initial Pressure Lb. per Sq. In. Gage	Back Pressure Lb. per Sq. In. Abs.	Superheat ° F.		
Locomotive single valve...	975	22 × 28	196.0	atmos.	....	55.6	23.4
Ames..........	248	17 × 16	100.0	atmos.	....	65.9	26.0
Buffalo Forge..	86	12 × 12	80.0	3.0	....	40.4	27.5
Corliss........	237	21.6 × 43.3	103.5	atmos.	....	79.5	11.7
Nordberg Poppet Valve.....	123	15 × 18	130.0	atmos.	....	80.0	13.4
Erie City Lentz	248	19 × 21 simple	133.0	atmos.	....	87.5	16.1
Worthington Pump Engine	648	Triple Exp.	146.8	0.78	87	72.5	10.0
McIntosh & Seymour......	2202	29 × 60 × 56	158.0	2.30	92	79.8	11.2

* From Tables 81 and 83 "Steam Power Plant Engineering," Gebhardt.

Cylinder Condensation. — This loss is the result of hot steam coming into contact with cooler surfaces, which absorb heat and produce condensation. The difference in the weight of steam as calculated from the indicator diagram and that actually used is caused mainly by cylinder condensation. This loss is the largest loss in a steam engine and varies from 16 to 20 per cent of the total loss. It is increased by using large clearances, and high ratios of expansion, because the volume of steam entering is then small in proportion to the volume of the cylinder and consequently a greater proportion of it is condensed.

Cylinder condensation may be reduced by using compound engines, unaflow and poppet-valve engines, by decreasing the clearance volume, increasing speed, decreasing ratio of expansion by making cut-off occur later, and by using superheated steam.

Leakage and Clearance. — The amount of leakage past the piston and valves is difficult to determine, but tests have shown that it may amount

to from 4 to 20 per cent. The greater the wear of the parts upon which the steam acts, the greater is the leakage. It is greater with saturated steam than with superheated steam. This loss is generally included as a part of the condensation loss.

The loss caused by clearance is mainly that resulting from increased condensation. The greater the proportion of the clearance volume to the total cylinder volume, the greater is the loss. The clearance volume of engines varies from 1 per cent for large engines having short steam passages to around 12 per cent for small high-speed engines. Clearance affects the economy because of its effect on the ratio of expansion. Shortening the cut-off increases the ratio of the clearance space to the volume of the steam supplied, and hence increases the loss.

Loss caused by Incomplete Expansion and Compression. — This loss is unpreventable, because the loss which results from having release occur before the expansion is complete is more than offset by reduced cylinder condensation since the cylinder walls are maintained at a higher temperature.

Compression is desirable to produce a smooth-running engine and, for a certain ratio of compression volume to clearance volume, means a saving in the amount of steam used.

Loss from Wire-drawing. — This loss is principally a loss in power resulting from the reduction in the pressure acting on the piston. The reduction in pressure is caused by restricted openings for the passage of steam. It is greatest in engines using single valves to control the distribution of steam. The use of multi-valve engines, such as Corliss and poppet valve, prevents this loss.

Loss from Friction. — The friction loss represents a loss in power, and varies from 4 to 20 per cent. It is larger when running under load than when under no load, because of the greater pressure on guides and bearings.

Loss from Moisture. — Unless a separator is used, the steam which condenses in the steam pipe passes through the engine as inert matter and reduces the work performed per pound of water and steam discharged. As this does not represent an actual loss, the performance of the engine is generally based on dry steam.

Loss from Heat Discharged in Exhaust Steam. — This loss is large and represents from 75 to 95 per cent of the heat supplied. Its amount depends upon the pressure against which the engine exhausts, an increase in the back pressure increasing the loss. In computing the quantity of heat in exhaust steam, the quality of the steam at exhaust must be known. This is a variable quantity, but tests have shown that its average value is approximately 90 per cent. Its value can be closely estimated by the following method:

$$\text{Heat in the exhaust steam, } H_2 = H_1 - H_r - \frac{2547}{W}$$

in which H_1 = total heat per pound of entering steam, B.t.u.

H_r = heat loss per pound of steam caused by radiation, B.t.u. This value may be assumed as 1 per cent of H_1 for most practical cases.

W = water rate per indicated horsepower.

$H_2 = x_2L_2 + h_2$

Substituting this value of H_2 in the above equation, there results

$$x_2 = \frac{H_1 - H_r - h_2 - \dfrac{2547}{W}}{L_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

Example 45. — The high-pressure cylinder of a triple-expansion engine develops 45.3 hp.; steam, dry and saturated; throttle pressure, 135.5 lb. per sq. in. abs.; exhaust pressure, 42.0 lb. per sq. in. abs.; weight of steam used per hr., 1626 lb. Find the quality of the exhaust steam.

Solution. — Using Equation (92),

$$x_2 = \frac{H_1 - H_r - h_2 - \dfrac{2547}{W}}{L_2} = \frac{1193.25 - 11.93 - 238.8 - \dfrac{2547}{35.9}}{933.5} = 0.935$$

H_1 at 135.5 lb. = 1193.25 B.t.u.; $H_r = H_1 \times .01$ = 11.93 B.t.u.; h_2 at 42 lb. = 238.8 B.t.u.; L_2 at 42 lb. = 933.5 B.t.u.; $W = \dfrac{1626}{45.3}$ = 35.9 lb. per i.hp. per hr.

405. Methods of improving Economy in the Steam Engine. — The following methods are used to increase the economy of steam engines:

1. Increasing boiler pressure and rotative speed.
2. Decreasing back pressure (Chapter XXVI).
3. Superheating the steam.
4. Steam-jacketing the cylinders (in some cases).
5. Compounding and using reheating receivers between cylinders (Chapter XX).
6. Using unaflow engines.

By **increasing the boiler pressure,** with other conditions remaining the same, a higher theoretical efficiency is obtained. The limit of the increase, at the present time, appears to be about 600 pounds per square inch. Leakage of steam and cylinder construction are the limiting factors. Present practice ranges from 90 pounds per square inch for low-pressure engines to 175 for unaflow condensing engines, 210 for locomotives, and 250 for marine engines.

An **increase in the rotative speed** of an engine does not always result in increased economy, much depending upon the type of the engine. High-speed engines, as a class, have the advantage of being more compact for a

given power, are simple in construction and are relatively low in first cost. They are, however, subject to quite rapid depreciation and, unless specially balanced, to excessive vibration, and are less economical in steam consumption.

The **saving effected by using superheated steam** is mainly a saving resulting from decreased initial condensation. The amount of steam used is reduced approximately 1 per cent for each 10 deg. fahr. increase in

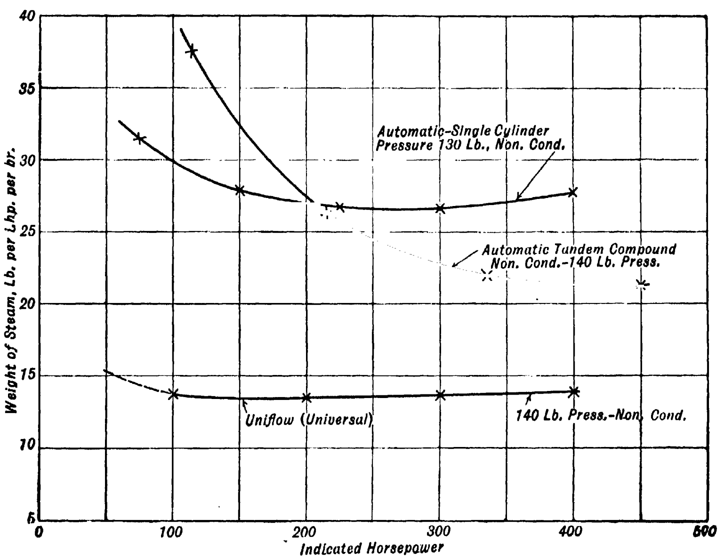


FIG. 320. — Water Rate Curve of Unaflow Engine compared with Two other Types of Engines.

superheat, the limit of the increase in superheat being about 670 deg. fahr. Superheated steam requires special valves, such as piston and poppet valves; if other valves are used, trouble will be experienced from warping and sticking of the valves.

The **use of steam jackets** generally reduces cylinder condensation. Their use is questionable, however, except for low-speed engines, working under steady loads, such as pumping engines.

The **use of a unaflow engine,** in which the admission valves and cylinder walls are not cooled by the exhaust steam, results in increased economy for all loads, because of reduced cylinder condensation. The economy of the unaflow engine is equal to that of a compound engine, and, because of its flat economy curve, Fig. 320, it can be used over a wide range of loads. Economy curves for a simple and a compound engine are also plotted in

Fig. 320, to give an idea of the difference in economy at light and heavy loads.

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Elements of Heat Power Engineering, HIRSHFELD AND BARNARD.
Thermodynamics, EMSWILER.

REVIEW QUESTIONS AND PROBLEMS

1. Describe the construction of a steam-engine indicator.

2. Give three uses of the steam-engine indicator.

3. State the proper size of indicator springs for steam pressures of 100, 150 and 200 lb. per sq. in.

4. Explain the method of placing a spring in the following indicators: (*a*) Crosby inside spring, (*b*) Thompson inside spring.

5. What is the purpose of a reducing motion? Name two types.

6. A reducing motion like that shown in Fig. 309, page 386, is attached to a 6 in. by 6 in. engine. Find the length of bar *fn* to give a diagram $2\frac{1}{2}$ in. long, if *af* equals 18 in.

7. Describe the proper method of taking an indicator diagram.

8. Develop the indicated-horsepower equation from fundamental considerations.

9. A Corliss engine provided with a Prony brake has the following dimensions: area of cylinder, 50.27 sq. in.; length of stroke, 18 in.; area of piston rod, 2.77 sq. in.; length of brake arm, 63 in. The following data were taken during a two hour test: barometer, 29.52 in. mercury; pressure at throttle, 118.5 lb. per sq. in. gage; pressure in exhaust main, 1 lb. per sq. in.; quality of steam, 0.99; net weight on brake scales, 250 lb.; weight of condensate, 1463 lb.

Area of diagrams: H. E., 3.00 sq. in.; C. E., 3.10 sq. in.

Length of diagrams: 4.00 in.; scale of spring, 80 lb.; r.p.m., 100.

Find: (*a*) i.hp., (*b*) b.hp., (*c*) mechanical efficiency.

10. It is desired to test an 8 in. by 24 in. Corliss engine at full load, which is 25 horsepower. The length of the brake arm is 7.86 ft.; r.p.m., 110; and the tare weight or the brake scales, 40 lb. Find the proper reading of the brake scales during the test.

11. Explain the method of using a planimeter to find an area.

12. An indicator diagram is 3 in. long and has an area of 2 sq. in. What is th mean effective pressure when each of the following springs are used: (*a*) 160 lb., (*b* 60 lb., (*c*) 20 lb.?

13. A 60-lb. spring was used to produce the diagram shown in Fig. 283, page 36 The clearance for each end was 6 per cent, and the atmospheric pressure 14.7 lb. per sc in. Find the water rate from the diagram.

14. Compare a theoretical indicator diagram with an actual diagram, stating th points of difference.

15. An 18 in. by 24 in. double-acting steam engine gives the following data: initi steam pressure, 110 lb. per sq. in. gage; quality of steam, dry; exhaust pressure, 2 ll per sq. in. gage; barometer, 28.5 in. mercury; cut-off, one-quarter stroke; r.p.m 120; diagram factor, 0.80.

Find: (*a*) theoretical m.e.p., (*b*) actual m.e.p., (*c*) probable i.hp.

16. Using the data of Example 9, find: (*a*) the thermal efficiency, (*b*) Rankin cycle ratio.

17. An 8 in. by 12 in. engine uses 500 lb. of steam per hour. Steam pressure, 110.3 lb. per sq. in. gage; quality 0.99; indicated horsepower, 21.5. Engine operating non-condensing with pressure of atmosphere, 14.7 lb. per sq. in.

Find: (*a*) thermal efficiency, (*b*) Rankine-cycle ratio.

18. Name the sources of heat loss in a steam engine. Which of these losses is the greatest?

19. Name three methods of increasing steam-engine economy.

CHAPTER XX

COMPOUND AND MULTIPLE EXPANSION ENGINES

406. Foreword. — Compound and multiple-expansion engines use steam at high pressure. After expanding and doing work in the first, or high-pressure, cylinder, the steam passes to one or more larger cylinders, where it does more work by expanding to a low pressure.

The principal object of compounding is to reduce the amount of steam used per horsepower per hour, by reducing the loss resulting from condensation of the entering steam. If the expansion of the steam takes place in two or more cylinders, the difference in pressure and, therefore the difference in temperature, in each cylinder is less than it would be if the whole expansion occurred in one cylinder. Consequently, the initial condensation of steam on the walls of the cylinder is reduced

The loss of heat by radiation, from compound engines, is somewhat larger than for a simple engine, on account of the larger surface of the cylinders; and, in addition, the friction losses may be greater because of the larger number of moving parts.

If the gain by decreased condensation so far exceeds the loss by increased radiation and friction, as to allow for increased cost, the use of the compound engine is justified. This generally occurs when the initial steam pressure is over 100 pounds per square inch Compound engines usually have the additional advantage of more uniform turning effort and smaller sizes of the separate parts

The **nominal size** of a compound engine is stated in terms of diameters and the stroke, beginning with the smaller cylinder. Thus the size of an engine having high- and low-pressure cylinders 15 and 24 inches in diameter respectively, and a stroke 48 inches long is written, 15 × 24 × 48 inches.

407. Classification. — Compound engines are classified in the following ways:

1. By position of cylinders
   - Center lines
     - Vertical
     - Horizontal or inclined
     - Horizontal and vertical
2. By arrangement of cylinders
   - Axis in same center line; tandem compound
   - Center lines parallel
     - duplex compound
     - cross compound
   - Center lines at right angles; angle compound

3. By number of cylinders	Two cylinders	duplex tandem cross-compound angle compound
	Three cylinders —	triple expansion, having high, intermediate, and low pressure cylinders
	Four cylinders —	quadruple expansion, having high, first and second intermediate, and low pressure cylinders

If two cylinders of a four-cylinder engine are low-pressure cylinders, the engine is called a four-cylinder, triple-expansion engine. Three-cylinder compound engines, like the inclined engines used on large side-wheel steamers, have one high-pressure and two low-pressure cylinders, one on each side of the high-pressure cylinder. A vertical compound engine is sometimes called "**a fore-and-aft**" compound, from its use on shipboard. An angle-compound engine has a horizontal high-pressure cylinder and a vertical low-pressure cylinder, the connecting rods bearing on the same crank pin.

408. Receivers. — If the cranks are at 90 degrees with each other, as in a cross-compound engine, a considerable part of the exhaust stroke of the high-pressure cylinder is still to be completed when cut-off comes in the low-pressure cylinder. For this reason, a receiver or storage vessel is provided to receive steam from the high-pressure cylinder and to furnish it to the low-pressure cylinder, as required, when the high-pressure cylinder is not exhausting. The volume of the receiver is ordinarily about twice the volume of the low pressure cylinder, and the pressure is such that the work performed in each cylinder is nearly equal.

If the engine has a single crank, as in the tandem arrangement, or if the cranks are at 180 degrees, steam passing from the high-pressure cylinder enters the low-pressure cylinder up to cut-off. This cut-off is made late, and steam then remaining to be exhausted from the high-pressure cylinder is compressed, up to the end of the stroke, in the pipe connections between the cylinders. Such engines are called non-receiver compounds, or "**Woolf**" engines.

Receivers may be used between the cylinders on triple- and quadruple-expansion engines. They often contain reheating coils into which live steam is passed for the purpose of adding heat to the working steam.

409. Tandem-compound Engines. — The tandem-compound engine, Fig. 321, has two cylinders, high and low pressure, arranged on the same center line. The piston rod of the high-pressure cylinder is attached to

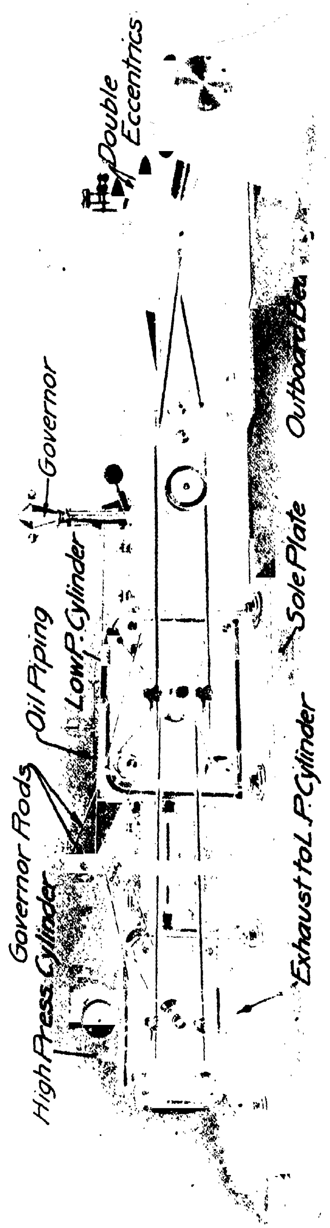

FIG. 321. — Tandem Compound Corliss Engine.

the low-pressure piston. The power developed in the high-pressure cylinder, plus that developed in the low-pressure cylinder, is transmitted by the low-pressure piston rod, crosshead, connecting rod, and crank to the shaft and flywheel.

Steam is admitted first to the high-pressure cylinder, where it performs work. During the admission and expansion period for the head end, steam is exhausted from the crank end through a short pipe connection to the proper side of the low-pressure piston to move it in the same direction that the high-pressure piston is moving. Thus, the back pressure in the high-pressure cylinder is the forward pressure in the opposite end of the low-pressure cylinder, except for a small loss in pressure which occurs in passing through the ports and pipe connections.

This type of engine occupies less floor space than other compound engines of equal capacity, but the cylinder nearest the shaft is less readily accessible for repairs.

410. Cross-compound Engines. — The cylinders of a cross-compound engine, Fig. 322, are placed side by side, the space between them allowing room for the valve gear, flywheel, and generator, in case one is used. Each cylin-

der has its piston, piston rod, crosshead, crank, and valve gear, which are proportioned in accordance with the work to be done in the cylinder.

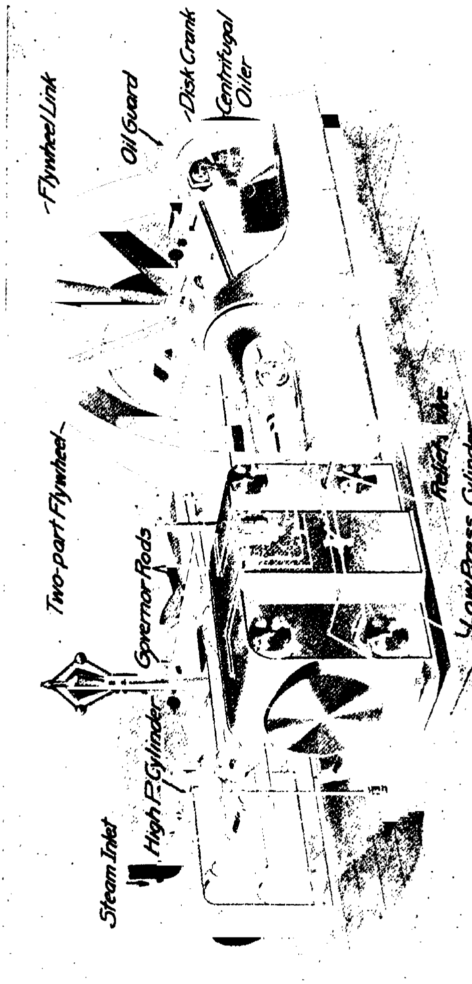

FIG. 322. — Heavy Duty Cross-compound Corliss Engine — Tangye Type.

The cranks are of the overhung type and generally are set at an angle of 90 degrees to each other.

This engine, with cranks at right angles, is usually provided with a receiver. The path of the steam is from steam supply to high-pressure cylinder, thence to receiver, to low-pressure cylinder, and to exhaust pipe under normal running conditions. When the engine is started, if the crank

pin of the high-pressure cylinder is at dead center, a by-pass in the pipe connections allows steam from the supply line to be admitted directly into the low-pressure cylinder. Thus the steam acts to advantage on the low-pressure piston, which is near the middle of its stroke while the crank is at 90 degrees with the line of centers. As in the tandem-compound engine, the length of stroke in both cylinders is made the same.

The space required for the cross-compound engine is larger than that occupied by the tandem engine of the same capacity. The parts which

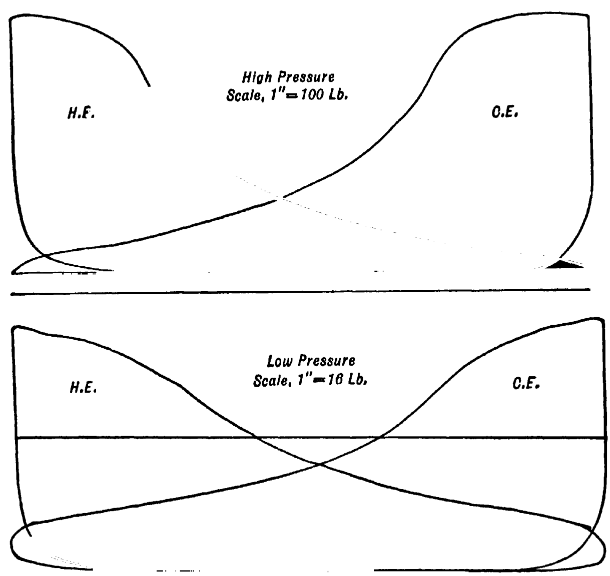

Fig. 323. — Indicator Diagrams from 28″ × 60″ × 60″ Cross-compound Engine.

transmit the power may be made lighter for the same total output, since the power is exerted through two separate sets of members. The turning force on the crank shaft is steadier, and, for the same uniformity in rotational effort, the flywheel may consequently be lighter than for a tandem engine.

Indicator diagrams from a 28 × 60 × 60 inch cross-compound engine are shown in Fig. 323.

411. Duplex-compound Engine. — This engine, Fig. 324, has the high pressure and low-pressure cylinders made in a single casting. The center lines of the cylinders are parallel, and the pistons and piston-rods are attached to a common crosshead. The reciprocating parts including the connecting rod and crank shaft, with counter-weights, are clearly shown. A piston valve is used.

412. Angle-compound Engines. — An angle-compound engine is shown in Fig. 325. It consists of a frame carrying a horizontal high-pressure

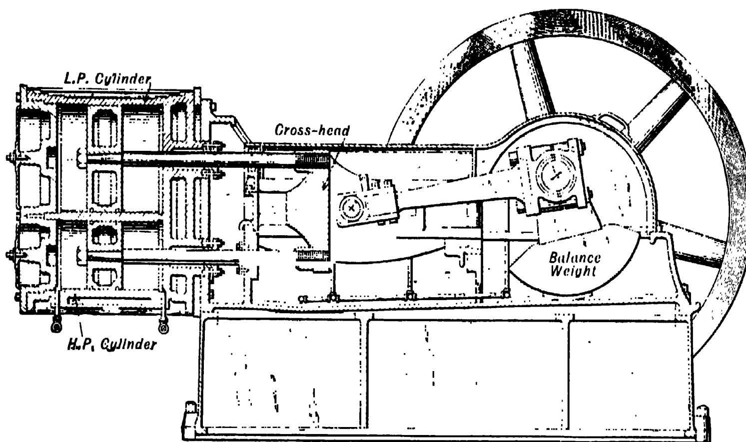

FIG. 324. — Section of Ball Duplex Compound Engine.

cylinder and a vertical low-pressure cylinder, together with their reciprocating parts, connecting rods, and a common crank-pin shaft and flywheel. The connecting rods are arranged side by side on a crank pin of double

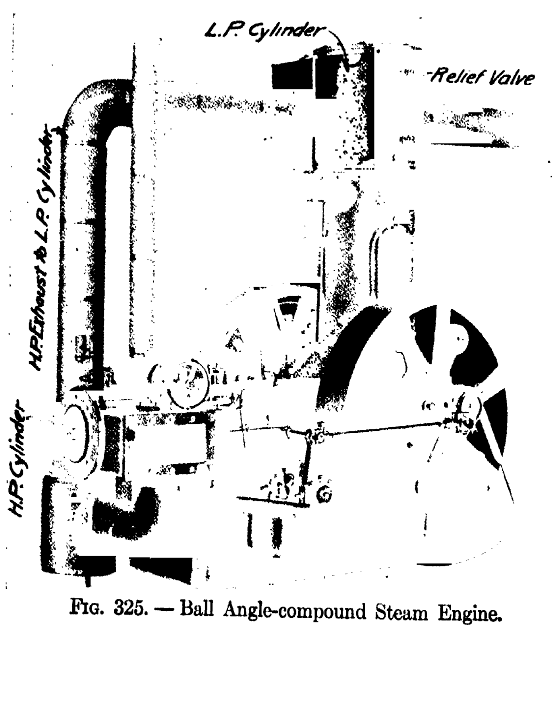

FIG. 325. — Ball Angle-compound Steam Engine.

the usual length. The counter-weight is placed directly opposite the crank pin, and is arranged to balance the inertia thrust of the horizontal reciprocating parts, when the crank pin passes the horizontal centers, and similarly to balance the vertical reciprocating parts when the crank pin passes the vertical dead centers, provided that both sets of moving parts are made to weigh the same. The engine is thus well balanced and may be run at speeds higher than ordinary. The **torque,** or the tendency of a force to turn a body about a point, is nearly constant, and a lighter flywheel can be used than with a simple engine of the same capacity. A shaft governor controls the high-pressure valve; the low-pressure valve is operated by a single eccentric. The pipe connections, with an enlargement of the pipe near the high-pressure cylinder furnish sufficient receiver space between the cylinders.

413. Cylinder Ratio and Expansion Ratio. — *The cylinder ratio for multiple expansion engines is the ratio of the piston displacement of the low-pressure cylinder to that of the high-pressure cylinder.* Since the stroke is made the same for all the cylinders, the volumes are to each other as the areas, or as the squares of the diameters.

For a compound engine, if C = cylinder ratio, L = stroke, d = diameter of the high-pressure cylinder, and D = diameter of low-pressure cylinder,

$$C = \frac{L\ \pi/4\ D^2}{L\ \pi/4\ d^2} = \frac{D^2}{d^2} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (93)$$

The ratio of expansion, or number of expansions of steam, R, for a compound engine, is the ratio of the volume of the low-pressure cylinder to the volume in the high-pressure cylinder at cut-off.

If r = number of expansions in the high-pressure cylinder, then $\frac{1}{r} \times L \times \frac{\pi d^2}{4}$ = volume at cut-off in high-pressure cylinder, and

$$R = L\frac{\pi D^2}{4} \div \left[\frac{1}{r} \times L \times \frac{\pi d^2}{4}\right] = r\frac{D^2}{d^2} = rC \quad . \ . \ . \ . \quad (94)$$

No account of clearance volume is taken in the above ratios.

Example 46. — A cross-compound engine has the following dimensions: high-pressure cylinder, 20 in. diameter; low-pressure cylinder, 40 in. diameter; stroke, 30 in. Cut-off in high-pressure cylinder at one-quarter stroke. Find the cylinder ratio and the ratio of expansion.

Solution. — Using Equations (93) and (94), with proper substitutions,

$$\text{Cylinder ratio} = C = \frac{D^2}{d^2} = \frac{1600}{400} = 4$$

$$\text{Ratio of expansion} = R = rC = 4 \times 4 = 16.$$

Modern compound condensing engines of the stationary type use steam at pressures from 125 to 175 pounds gage. Triple-expansion marine

engines generally use steam at 150 to 200 pounds gage, and quadruple-expansion marine engines at 175 to 225 pounds gage.

The cylinder volumes corresponding to these pressures range approximately as in the following table:

TABLE 30. — CYLINDER RATIOS FOR COMPOUND AND MULTI-EXPANSION ENGINES

Kind of Engine	Range of Ratio		Order of Cylinders in which Ratio is expressed
	From	To	
Compound..........	1 to 4.2	1 to 4.6	High to low
Triple expansion.....	1 to 2.3 to 6	1 to 2.5 to 7.8	High to intermediate to low
Quadruple expansion.	1 to 1.8 to 3.6 to 7.8	....	High to 1st intermediate to 2nd intermediate to low

Non-condensing engines using pressures from 100 pounds to 125 pounds have cylinder ratios from 2.5 to 3.25.

The tendency is toward much larger cylinder ratios for compound engines, tests having shown good economy with ratios as high as 7.0 to 1.0.

Cut-off in the high-pressure cylinder of stationary engines ranges from 0.25 to 0.4 stroke, under normal load conditions, and from 0.55 to 0.75 stroke for marine engines.

Triple-expansion engines are used in stationary practice, where the load is steady, as in pumping stations.

414. Indicated Horsepower. — *The horsepower of a compound or multiple-expansion engine is the sum of the horsepowers developed simultaneously in the cylinders.* The power for each cylinder may be found from the indicator diagrams, as explained in Art. 390, page 388.

Example 47. — A $20 \times 40 \times 30$ in. vertical cross-compound engine, with both piston rods 4 in. in diameter, running at 145 r.p.m., gave data from indicator diagrams as follows: Scale of indicator springs; high pressure, 80 lb.; low pressure, 16 lb.

MEAN EFFECTIVE PRESSURE

High pressure	Area	Length	M.e.p.	*Low pressure*	Area	Length	M.e.p.
H.E..........	2.36	3.95	47.80	H.E.........	2.72	4.05	10.74
C.E..........	2.89	3.95	58.53	C.E..........	2.93	4.05	11.57

PISTON AREAS

H.E. area = 314.16 sq. in.	H.E. area = 1256.64 sq. in.
C.E. area = 301.59 sq. in.	C.E. area = 1244.07 sq. in.

Find the i.hp. of the engine.

Solution. — Using Equation (82), page 388, and the data as given, the horsepower of the engine is found as follows:

High-pressure cylinder

$$\begin{cases} \text{H.E. i.hp.} = \dfrac{PLAN}{33{,}000} = \dfrac{47.80 \times 2.5 \times 314.16 \times 145}{33{,}000} = 165 \\ \text{C.E. i.hp.} = \dfrac{PLAN}{33{,}000} = \dfrac{58.53 \times 2.5 \times 301.59 \times 145}{33{,}000} = \underline{194} \end{cases}$$

359

Low-pressure cylinder

$$\begin{cases} \text{H.E. i.hp.} = \dfrac{PLAN}{33{,}000} = \dfrac{10.74 \times 2.5 \times 1256.6 \times 145}{33{,}000} = 148 \\ \text{C.E. i.hp.} = \dfrac{PLAN}{33{,}000} = \dfrac{11.57 \times 2.5 \times 1244 \times 145}{33{,}000} = \underline{158} \end{cases}$$

306

Total horsepower for the engine.. 665

415. Rated Horsepower. — The probable, or rated, horsepower can be calculated from the dimensions of the engine and the stated conditions, by considering that all the power could be developed in the low-pressure cylinder, except for practical reasons. In the actual engine, the same weight of steam, on the average, is exhausted per stroke from the low-pressure cylinder as is admitted to the high-pressure cylinder, and consequently the work may be figured as though it were all done in the low-pressure cylinder, with the total number of expansions equal to R. A suitable diagram factor is applied, to find the mean effective pressure. Thus, if

N = revolutions per minute.
P_1 = initial pressure, lbs. per sq. in., absolute.
P_3 = back pressure for low-pressure cylinder, lbs. per sq. in., absolute.
f = diagram factor.
C = cylinder ratio, and $R = Cr$.

Cut-off in high pressure cylinder at $\frac{1}{r}$ part of the stroke.

$$\text{M.e.p.} = f\left(\frac{P_1(1 + \log_e R)}{R} - P_3\right) \text{lb. per sq. in.}$$

Rated horsepower for a double-acting engine =

$$\frac{\text{m.e.p. (lb.)} \times L \text{ (ft.)} \times \text{Area } L.P. \text{ cylinder (sq. in.)} \times 2 \times N}{33{,}000} \quad (95)$$

The rated horsepower for a stationary engine is generally calculated with cut-off in the high-pressure cylinder at one-quarter stroke.

416. Diagram Factors for Compound Engines. — Slide-valve engines, 0.55 to 0.80; Corliss engines, 0.75 to 0.85; triple-expansion engines, 0.55 to 0.70.

417. Governing Compound Engines. — The compound engine is generally governed (1) by controlling the length of admission of steam into the high-pressure cylinder, with cut-off on the low-pressure cylinder fixed in one position, or (2) by simultaneous control of cut-off in both cylinders.

The governor may be of the loaded-pendulum type, or of the centrifugal-inertia type. The total work done by the engine is proportional to the

amount of steam admitted into the high-pressure cylinder. *The distribution of work between the cylinders is controlled by the cut-off in the low pressure cylinder and is generally such that the total work is equally distributed between the cylinders.*

In case (1), with fixed low-pressure cut-off, the work is evenly divided for only one set of conditions. In case (2), the distribution of work is

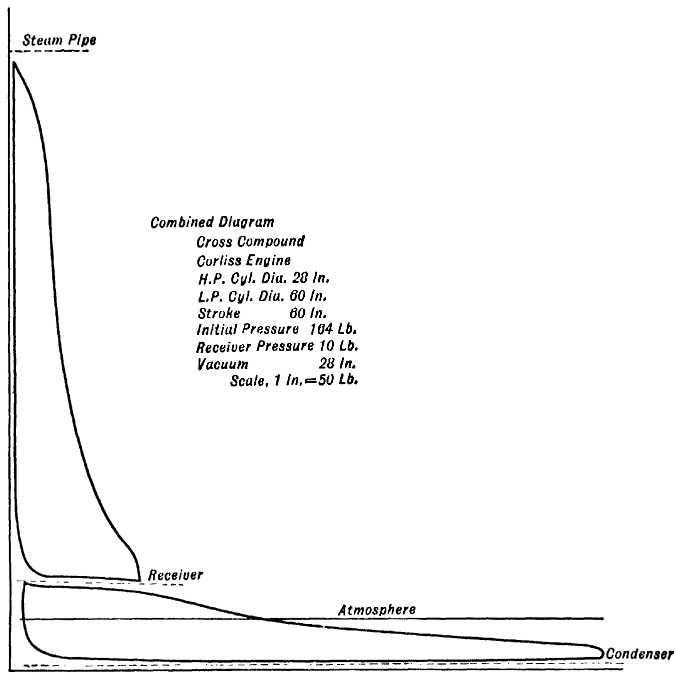

FIG. 326. — Combined Indicator Diagram made from Diagrams shown in Fig. 323.

controlled by low-pressure cut-off which is under control of the governor at all loads. It may be adjusted by hand while the engine is running. Early cut-off in the low-pressure cylinder increases the receiver pressure, since a smaller volume of steam is withdrawn than for a late cut-off. This increases the back pressure in the high-pressure cylinder and reduces the work done there, thus increasing the work done in the low-pressure cylinder if the total work remains the same. Therefore, in adjusting the valve gear to equalize the loads, if more work is to be done in the low-pressure cylinder the governor-rod yoke is adjusted to give earlier cut-off in that cylinder, thus increasing the receiver pressure. Later cut-off in the low-pressure cylinder reduces the receiver pressure, so that more work is done in the high- and less in the low-pressure cylinder. The receiver pres-

sure for a condensing engine, with initial pressure 160 pound gage, and 27-inch vacuum, is about 11 pound gage.

418. Combined Diagrams. — Indicator diagrams taken from the cylinders of a multiple-expansion engine may be redrawn to a common scale of pressures and volumes, giving what is called the "combined diagram." This diagram, Fig. 326, shows the change in pressure of the steam and its change in volume, from the time of its entrance to the high-pressure cylinder until it leaves the low-pressure cylinder. It reveals the drop in pressure between the cylinders and shows whether or not the receiver and pipe connections between the cylinders have sufficient capacity for the steam present when the diagrams were taken.

The combined diagrams for a compound engine may be laid out as follows: Redraw the low-pressure diagram, Fig. 323, using a convenient number of points, to the same length but with a pressure scale suitable for both diagrams. Lay out the zero of pressure axis, or axis of volumes, at a distance below the atmospheric line equal to the barometric pressure to scale, and the zero of volume axis at a distance from the end of the diagram equal to the per cent of clearance volume in the low-pressure cylinder times the length of the low-pressure diagram. The length of the high-pressure diagram, to the same scale as the low-pressure diagram, is found by dividing the length of the latter by the cylinder ratio. The high-pressure diagram is laid out, beginning at a point distant from the zero of volume axis by an amount equal to the per cent of clearance volume in the high-pressure cylinder times the length of the high-pressure diagram, to the reduced scale.

419. Economy of Compound Engines. — Standard-type compound engines, having a single valve for each cylinder, generally require from 22 to 27 pounds of saturated steam per i.hp. per hour, non-condensing. It should be stated that compound engines are seldom run non-condensing and then usually for small sizes. With a standard vacuum of 26 inches of mercury, the water rate is decreased about 20 per cent.

For four-valve engines, the water rate with saturated steam may be from 17 to 22 pounds per i.hp. per hour, non-condensing. When operating condensing, the water rate may be 12 to 14 pounds per i.hp. per hour, depending upon the initial conditions of the steam and the vacuum.

For saturated steam, the triple-expansion Allis Corliss pumping engine at Chestnut Hill, Boston, has a record of 10.33 pounds of steam per i.hp. per hour, corresponding to 196 B.t.u. per minute.

Using steam at high pressure and superheat, compound condensing engines having poppet valves, unaflow engines, and locomobile engines have shown better economy — in some exceptional cases, as low as 6 to 8 pounds per i.hp.-hour.

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Catalogues of Builders.

REVIEW QUESTIONS AND PROBLEMS

1. What is the principal reason for compounding steam engines?

2. How may compound engines be classified?

3. Why are receivers used?

4. Describe concisely the arrangement of the tandem-compound engine. What advantage has it over other compound engines?

5. Answer for the cross-compound engine as in Question 4.

6. Answer for the angle-compound engine as in Question 4.

7. What is "cylinder ratio," and what is ratio of expansion?

8. A double-acting cross-compound steam engine, $12 \times 22 \times 30$ in. has cut-off at quarter stroke in the high-pressure cylinder. What is the cylinder ratio? What is the rate of expansion?

9. Using data in Problem 8, with initial steam pressure, 125.3 lb. gage; back pressure, 4 lb. abs.; r.p.m., 130; diagram factor, 0.75; find the probable horsepower.

10. Given the following data from a test on a $28 \times 52 \times 48$ in. Corliss cross-compound condensing engine, calculate the i.hp.:

Indicator diagrams: High-pressure, head-end area 1.33 sq. in., length 3.89 in.; crank-end area 1.47 sq. in., length 3.89 in.; scale of spring, 80 lb. Low-pressure, head-end area, 1.46 sq. in., length 3.77 in.; crank-end area, 1.52 sq. in., length, 3.77 in.; scale of spring, 20 lb. Diameter of both piston rods, 6 in.; r.p.m., 82; at switchboard amperes, 600; volts, 615.

CHAPTER XXI

METHODS OF LUBRICATION
ENGINE ACCESSORIES

420. Foreword. — The rubbing surfaces of metal bearings are never perfectly smooth; even when they are highly polished, small depressions and elevations can be seen under a microscope. Under pressure, these depressions and elevations interlock and resist the moving force, and the movement of such contact surfaces, relative to each other, produces friction and wear. Friction produces heat, which is a form of energy and in this case represents a loss of work, the amount of this loss varying from 5 to 20 per cent of the total power developed by the engine. The distribution of the friction loss in a 6 inch by 12 inch straight-line engine, having an unbalanced slide valve, is approximately as follows: main bearing, 35.4 per cent; piston and rod, 25 per cent; crank pin, 5.1 per cent; crosshead and wrist pin 4.1 per cent; valve and rod, 26.4 per cent; eccentric strap, 4.0 per cent.

To reduce the loss caused by friction, a lubricant is placed between the surfaces. If the supply of lubricant is sufficient, an oil film is formed between the surfaces, and the resulting friction is merely fluid friction, or resistance of the molecules of the fluid to motion. The loss from fluid friction is much less than that resulting from poorly lubricated surfaces in which the oil film is not formed.

It would be impossible to operate machinery without lubrication, as the frictional heat developed would soon destroy the bearing surfaces.

The selection of the proper lubricant may mean a large increase in the economy of the engine. This selection is an important problem, which calls for a knowledge of the characteristics of lubricants and the best methods of applying them.

Lubricants used for engines may be classified as solid, such as graphite, semi-solid, such as heavy greases, and liquid. The most common liquid lubricant is mineral oil.

421. Characteristics of Oil. — An oil, to be efficient as a lubricant, should have the following characteristics:

1. Sufficient body, or combined capillarity and viscosity, to form an oil film between the bearing surfaces.
2. Least fluid friction compatible with sufficient body.

3. Freedom from tendency to decompose, or change in composition by gumming, on exposure to air or in use.

4. Absence of acids or properties liable to injure metal with which it comes in contact.

5. High temperature of vaporization, and low temperature of solidification.

6. Freedom from grit and foreign matter.

In the selection of a suitable lubricant, the points to be considered are its cost, its efficiency in reducing friction, and its durability under wear. Some of the best lubricating oils are too expensive to permit their general use as lubricants while many cheap oils have small lubricating power; other oils have all the desired qualities, but they soon wear out and lose their lubricating power. Some oils, such as linseed oil, are liable to gum so seriously as to prohibit their use as lubricants; while many lubricants contain injurious acids which have been used as clarifiers and have not been completely removed. Many lubricants, because of their low boiling-points, cannot be used in steam cylinders, and others are liable to congeal when subjected to low temperatures. The **viscosity** or the property of an oil that determines its rate of flow, is perhaps the most important quality, as it is a measure of the ability of the oil to form and maintain an oil film. The viscosity required depends upon the load to be carried, the temperature, and the speed. For light loads and high speeds, a light-body oil should be used, and for heavy loads and slow speeds a heavy-body oil. In any case, the body should be such as to form an oil film and still keep the fluid friction at a minimum. In general, the price of an oil is of small importance in comparison with its lubricating properties.

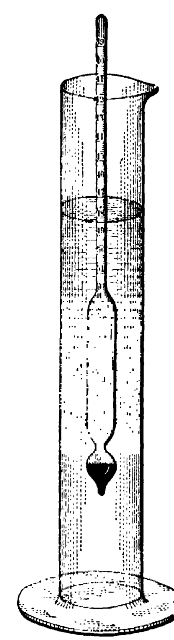

Fig. 327. — Hydrometer and Jar.

422. Methods of Testing Oil. — To determine the physical characteristics of an oil the following tests are made:

Specific Gravity Test. — *The specific gravity of oil is taken as its weight compared with that of the same quantity of water at 60 deg. fahr.* To determine the specific gravity of liquids lighter than water, a **hydrometer,** Fig. 327, is used. It has a weighted glass bulb with a stem that is graduated to read in degrees **Baumé,** when used for oil. The greater the Baumé reading, the lower is the specific gravity. Degrees Baumé are changed to specific gravity as follows:

$$\text{Specific gravity at 63.5 deg. fahr.} = \frac{140}{130 \times \text{deg. Baumé}}$$

In making a test, the hydrometer is placed in a jar of the oil and floats in a vertical position at a depth depending upon the density of the oil. The specific gravity is determined by the depth to which the hydrometer sinks, as shown by the markings on the stem with the oil at 63.5 deg. fahr.

The gravity test determines the uniformity in weight of an oil, but does not give any reliable conclusion regarding its lubricating value.

The **cold test** is made to determine the temperature at which oil ceases to flow. It must be carefully made or the results will not be satisfactory. A bottle of oil is placed in a freezing mixture, and, by frequent inspection of a thermometer fitted through the cork of the bottle, the temperature at which the oil congeals is obtained.

The **pour test** is made by freezing the oil in a partly filled bottle. The bottle is then removed from the freezing mixture and the temperature at which the oil will flow from one end of the bottle to the other is noted.

The **flash-point test** *indicates the lowest temperature at which vapor from an oil will ignite, but not continue to burn, when an open flame is brought near its surface.* It is made by applying heat to the oil, placed in a cup, and raising its temperature to the point of vaporization.

The **fire test** is a continuation of the flash point test and indicates the temperature at which the vapor of the oil will continue to burn. Its value is small as regards the value of a lubricant for bearings, because bearing temperatures rarely exceed 140 deg. fahr., while the lightest oils have flash points of over 300 deg. fahr.

423. Viscosity Test. — The Saybolt Universal Viscosimeter is used in the United States for measuring viscosity. By its use *the viscosity is determined as the time in seconds required for a known quantity of oil, at a definite temperature, to flow through an orifice of a known size.*

The oil to be tested is placed in the standard oil tube, at the bottom of, which is a small standardized outlet. The standard tube is surrounded by water or oil at the test temperature When the oil inside the tube reaches the test temperature, which is generally from 104 to 140 deg. fahr., since these are the bearing temperatures in steam engines, the plug at the bottom of the tube is removed, and the oil is allowed to flow into a glass flask of 60 cubic centimeters capacity. The time required to fill the flask is obtained by using a **stop watch.**

424. Methods of Lubrication. — The surfaces requiring lubrication in an engine may be divided into sliding and rotating surfaces. Flat sliding surfaces, such as the crosshead shoes, tend to scrape the oil from the guide, unless the corners are slightly rounded to prevent it; in general, such surfaces are imperfectly lubricated. Rotating surfaces, such as the journal in its bearing, tend to draw the oil in between the surfaces, because of the

viscosity of the oil. Bearings are ordinarily made with a clearance amounting to 0.001 inch per inch of diameter, which permits the oil to form a film, when supplied in sufficient amounts. Bearing surfaces are generally grooved to assist in distributing the oil. Sliding surfaces, such as that of the piston in the cylinder, are more difficult to lubricate.

The methods used to lubricate steam-engine bearings may be classified as follows:

1. For individual bearings	Hand oiling Drop-feed oiling Ring or chain Wiper Grease cup Oily waste
2. For group bearings	Splash system Gravity circulating system
3. Cylinder lubrication	Hydrostatic lubricator Mechanically operated lubricator

The methods listed in group 1 give imperfect lubrication, as the oil is supplied in limited amounts. Ring or chain oiling where applicable, is the best of these methods. Oil-bath and forced-feed methods give nearly perfect lubrication, since the oil is supplied in a sufficient amount to maintain an oil film.

Hand oiling is used for small bearings in which the rubbing speed is low, such as the moving parts of the valve gear. Oil is fed into an oil hole in the bearing cap, by an oil can having a flexible bottom. It then spreads and gradually works its way to the ends of the bearing. The oil hole is often covered with an oil cup having a spring-actuated cap which closes and keeps dust from getting into the bearing

Drop-feed oiling consists in feeding a fairly regular supply of oil to the bearing. The devices used for drop-feed oiling are the wick-feed oiler, generally used to lubricate the main and thrust bearings of marine engines, and the sight-feed oiler.

The **wick-feed oiler** has an oil chamber covered by a cap, a siphon oil tube projecting above the oil level, and a wick made of strands of woolen yarn. The wick dips into the oil and has its longer end projecting into the tube. Oil is drawn from the oil chamber by capillary action and drops on the shaft.

The **sight-feed oiler,** Fig. 328, is the most common type of oiler, and is easily regulated to feed the desired quantity of oil. The bottom of the **glass oil chamber,** which is used to show the height of oil, is made of brass and contains a conical seat at the bottom of a central sleeve. An **adjusting**

needle valve screws into the top of the cap, which is provided with a hole for filling. By turning the **milled collar,** the position of the needle is changed with respect to the conical seat, and the rate of feed regulated.

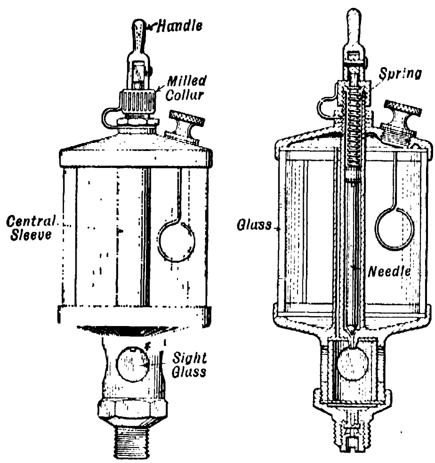

FIG. 328. — Sight Feed Oiler.

With the small handle attached to the top of the needle horizontal, a spring inside the tube pushes the needle against the conical seat and stops the flow of oil. When it is vertical the oil will feed as regulated. The rate of flow is observed through a sight hole placed in the base.

Sight-feed drop oilers are often arranged with multiple feeds, as shown in Fig. 354, page 455, in which there are seven oil feeders controlled by seven different needle valves located in oil tubes that deliver the oil to the various bearings.

The crank pins of overhung cranks are often oiled by a special arrangement, Fig. 329, using drop-feed oilers to regulate the rate of flow. The oil tube, bent to form a right angle, has one end attached to the crank pin and

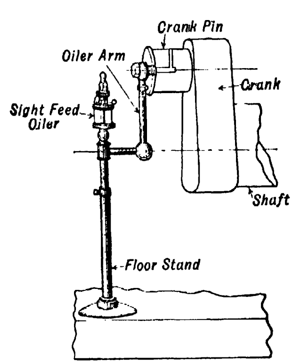

FIG. 329. — Centrifugal Crank Pin Oiler.

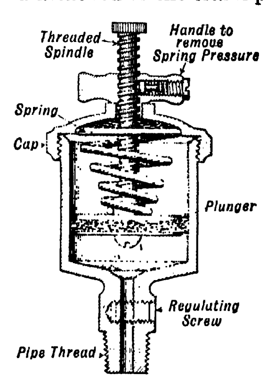

FIG. 330. — Grease Cup Oiler.

the other end enlarged to form a spherical chamber. The tube is of such length that the chamber is in line with the axis of the main shaft. Oil flows from the sight-feed drop oiler, usually fixed to a railing or standard,

through a feed tube, and drops into the chamber, from which it is delivered to the crank pin by centrifugal force and enters the bearing through holes drilled in the crank pin, as shown.

A **telescopic oiler** is used on many engines to lubricate the crosshead pin. A sight-feed oil cup delivers oil to a tube, which slides up and down in a tube attached to the crosshead. Oil flows from the oil cup through the tubes to the wrist pin, and then through drilled holes to the bearing.

An **oil wiper** is often used to lubricate the crosshead pin of slow-running engines. A wick, lubricated from a drop oil cup, hangs in the path of the wiper. The wiper strikes the wick and the oil thereon drops into the body of the oiler and flows to the wrist pin bearing, through holes drilled in the wrist pin.

A **grease cup,** Fig. 330, is used to lubricate eccentrics and other slow-moving parts. Grease is placed in the cup below the cup piston, where it is under a pressure produced by a light spring. When the bearing becomes sufficiently warm, some of the grease is melted and lubricates the bearing. When not in use, the small flanged nut raises the piston and removes the pressure from the grease.

Ring and chain oiling are used to lubricate the main bearings of high-speed engines. The bearing housing forms an oil reservoir in which oil is maintained at a level sufficiently high to have the lower ends of the rings submerged. One or two rings or chains surround the shaft and dip into the oil. The rings revolve with the motion of the shaft and carry oil to the top of the shaft. The oil is then distributed to the bearing by the oil grooves. The surplus oil flows to the ends of the bearing and drops back into the oil reservoir, where it is sometimes cooled by water circulating through coils of piping.

Splash oiling is employed to lubricate a number of bearings in an enclosed casing. It is used extensively with small vertical and horizontal high-speed engines. The enclosed crank chamber, Fig. 331, contains oil at such a level that the crank pin dips into the oil at each revolution. The level of the oil is maintained constant by using an overflow connection. The crank disk, revolving inside the crank chamber, dips into the oil, and thus oil is picked up, and is thrown off the revolving rim by centrifugal force. Oil wells or **pockets,** cast on the inside of the casing, collect the oil and lead it through various channels, tubes, or troughs to the parts requiring lubrication. The main bearings and eccentrics, crank pin, crosshead pin, and crosshead guides are lubricated either directly by means of oil spray or indirectly by means of the oil troughs.

425. Circulation Oiling System. — There are two systems embodying the oil circulation principle:

1. Non-pressure oil circulating system.
2. Pressure oil circulating system.

426. Non-pressure Oil Circulating System. — By this system the oil delivered to the bearings is not under direct pressure. It is employed to lubricate automatically the main bearings, crank pins, crossheads, cross-

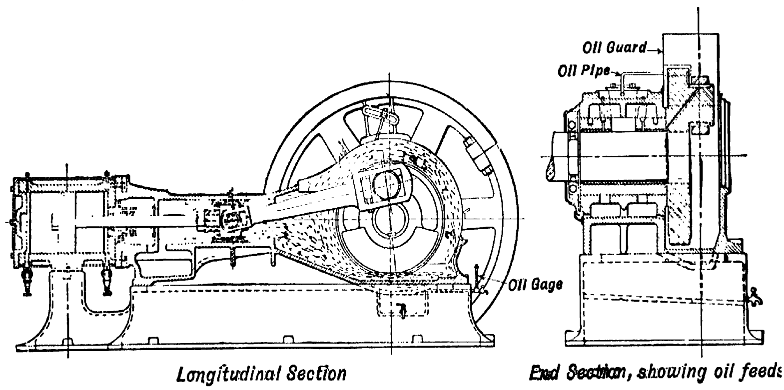

Fig. 331. — Splash System of Oiling.

head guides; or, in general, most of the external moving parts in medium and large sized steam engines.

A typical system is shown in Fig. 332. Oil flows by gravity from the **supply tank,** through distributing pipes, to sight-feed glasses in each pipe

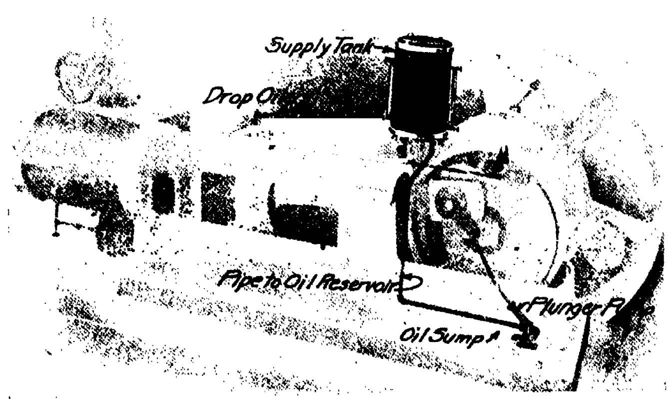

Fig. 332. — Non-pressure Oil Circulating System.

line, and the amount flowing to each bearing is regulated by needle valves at the sight-feed glasses. From the bearings, the oil drains through return oil pipes back into the **sump tank,** or lower part of the bed. An **oil pump,** driven by the engine, pumps the oil from the sump, either through an oil cooler or directly into the supply tank, which is provided with an overflow pipe to return the surplus oil to the sump tank.

427. Pressure Oil Circulation System. — Oil is delivered by this system, as directly as possible, to the various bearing surfaces requiring lubrication. It is used on enclosed types of steam engines and turbines. For description see Art. 478, page 492.

428. Lubrication of Internal Parts of the Steam Engine. — The internal parts of the steam engine, such as the valve, valve rod, piston, and piston rod, are exposed to steam temperature conditions. The valve and piston are not in view, and the condition of lubrication is not easily determined. For this reason, correct lubrication of these parts is more difficult to accomplish than that of the external moving parts.

Experience has shown that only by the use of a correct grade of high-quality cylinder oil, especially selected to suit the operating conditions of the engine and applied in the correct manner, to the right place, and in the proper quantity, will the valves and pistons of steam engines operate at highest efficiency with minimum wear. Only sufficient oil should be used to ensure an unbroken oil film.

Cylinder oil is an oil having a heavy body. *It is ordinarily introduced by a lubricator into the steam line at a point just above the stop valve. The oil is broken up and carried by the steam to the internal parts.* The following types of lubricators are used:

1. Hydrostatic lubricator.
2. Mechanically operated lubricator.

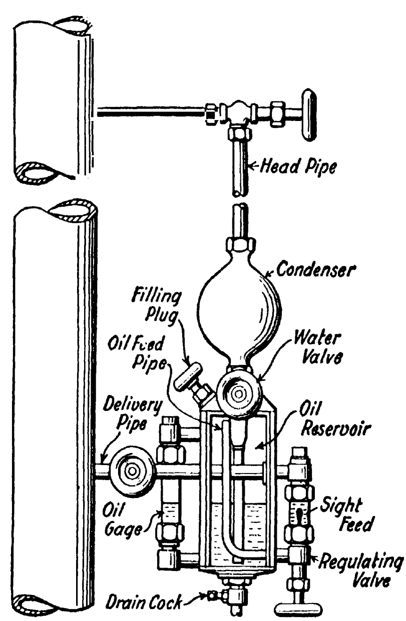

FIG. 333. — Hydrostatic Cylinder Lubricator.

429. Hydrostatic Lubricator. — This lubricator, Fig. 333, is widely used because of its simplicity. It consists of a chamber having two connections to the steam pipe leading to the engine. The chamber is filled with oil and water the relative amounts of which are shown by a gage glass. At the bottom of the chamber is a drain cock for draining off the water before refilling the lubricator through the **filling plug.** *When refilling, the valves in both steam connections to the lubricator should be closed, to prevent oil from being blown out when the filling plug is removed, and causing*

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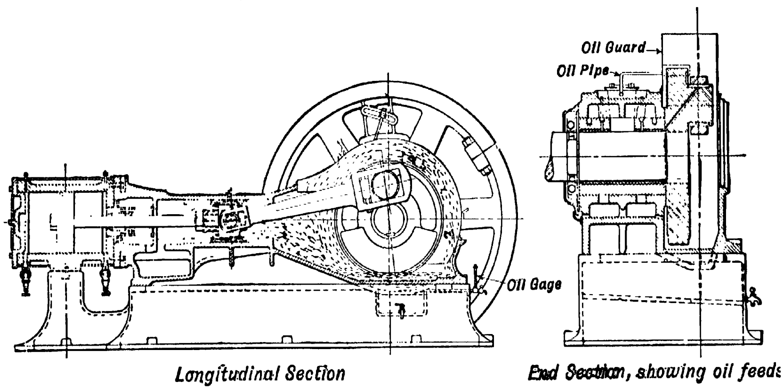

FIG. 331. — Splash System of Oiling.

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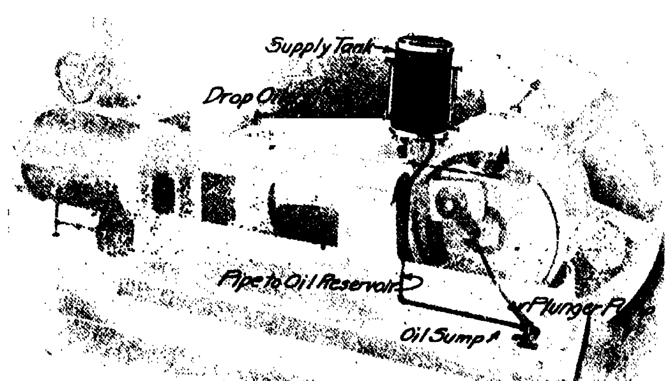

FIG. 332. — Non-pressure Oil Circulating System.

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two upper ball valves and through a **nozzle.** Oil-drops form around the guide wire and rise through the water in the sight-feed. The oil then passes a non-return valve and is forced through the check valve to the **atomizer** in the steam pipe. The rate of feed is controlled by two adjusting nuts located below the sight-feed.

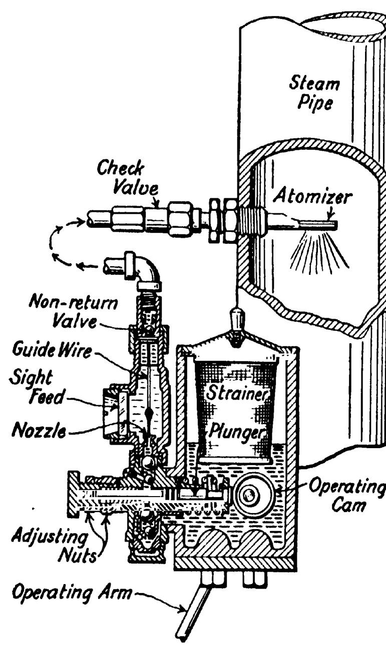

FIG. 334. — Positive Pressure Cylinder Lubricator.

The best method of distributing cylinder oil to the cylinder and valves is the atomizing method. Oil is introduced through an atomizer, which is spoon-shaped on its upper side and has slots extending through it, into the steam flowing to the cylinder. Steam at high velocity impinges against the upper surface of the atomizer and forces the oil through the slots. This breaks the oil up into a fine spray which enters the cylinder with the steam.

All types of engine valves are efficiently lubricated by this method, and the piston rod is also lubricated in a most economical manner. In multi-cylinder engines, each cylinder after the first is lubricated by oil carried by the exhaust steam from the preceding cylinder.

In vertical steam engines, less oil is required for the piston and cylinder than for horizontal engines, because there is less pressure between the piston, the piston rings and the walls of the cylinder.

In horizontal steam engines the cylinder takes the pressure caused by the weight of the piston sliding on the cylinder, in addition to the pressure caused by the piston rings. The extra pressure produced by the piston is sometimes removed by a **tail rod,** an extension of the piston rod passing out through the rear head and connected to a tail-rod support. The weight of the piston is then carried by the crosshead and tail-rod guides.

431. Effect of Oil in Exhaust Steam. — A part of the oil used to lubricate the internal parts of the steam engine passes out with the exhaust steam and is present in one of two forms: (1) oil in **suspension,** or (2) oil in **emulsion.** **Oil in suspension** is in the form of globules that can be removed from the

steam by an exhaust-steam oil separator. The globules of oil which are not extracted from the steam in the separator in the exhaust pipe line will, in the case of condensing engines, mix with the condensed steam and pass to the hot well, where the greater portion will rise to the surface and can be skimmed off. Filters using sand, wood or wool may be used to retain the globules, in case the pump suction is not taken low enough in the hot well to avoid drawing in the "float" oil. **Oil in emulsion** consists of very minute particles of water coated with an oil film. The particles, which are so small that the exhaust-steam separator will remove only a portion of them, may be removed by chemical precipitation or electrical treatment.

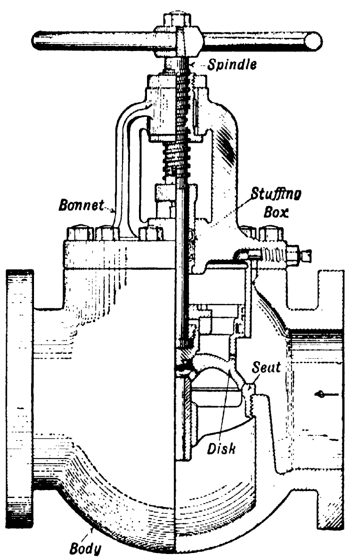

FIG. 335. — Steam Engine Throttle Valve.

Oil in emulsion has a milky appearance, due to the mixing of the oil with the steam. With only a small percentage of oil present in emulsion, the water is almost clear. Oil in emulsion will not rise to the surface in the hot well, and cannot be removed by feedwater filters.

When in this form, the oil combines with the impurities in the feedwater and coats the heating surfaces of the boiler, thus retarding the flow of heat and causing damage, and often failure of the boiler by overheating.

A **splash guard,** made of sheet metal, is used to prevent oil from being thrown from the revolving crank pin and connecting rod end over the floor and walls of the building. The guard collects the escaping oil and drains it into the base of the frame. For a typical installation see Fig. 322, page 413.

432. Engine Accessories. — The **throttle valve,** Fig. 335, is placed near the engine cylinder to control the flow of steam to the engine. It consists of a cast-iron body having two flanges. One is bolted to the engine cylinder and the other to a flange on the main steam line. A valve seat is machined in the body of the valve, upon which a valve disk acts. The position of the disk is controlled by a spindle which passes through a stuffing box. The threaded portion of the spindle is supported by an **outside yoke,** and carries a handwheel by which the valve is opened and closed.

The cylinders of steam engines are generally protected, by a relief valve or an explosion diaphragm, against dangerous rise in pressure caused by

water from an undrained pipe or foaming boiler. The **relief valve** is a small safety valve which opens to let the water escape from the cylinder. The **explosion diaphragm** is a cast-iron disk having an accurately pre-determined section which will give way and release the water as soon as the pressure rises 50 per cent above initial steam pressure. Hand-operated valves are provided to shut off the diaphragms when necessary

433. Steam Separator. — *The steam separator removes the entrained water, which collects at the bottom of a steam pipe and which is the result of moisture in the steam when it leaves the boiler and of condensation in the steam line.* Unless this water is removed from the steam it may be carried into the cylinder, be caught between the piston and cylinder head, and result in

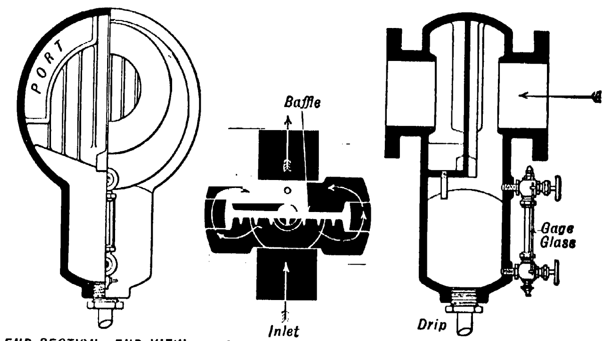

FIG. 336. — Cochrane Baffle-type Separator.

breaking the cylinder head; if carried into a turbine it may pit and wear the blading. The danger from water in steam is increased when the velocity or pressure of the steam is increased. Steam turbines, reciprocating engines, steam pumps, or other equipment utilizing live steam should be protected by separators. The economy of engines and turbines is increased by decreasing the moisture content of the steam.

Steam separators may be classified as:

1. Reverse-current separators, in which the direction of flow is abruptly changed, usually through 180 degrees. The water in the steam, because of its greater weight and inertia, is thrown into the receiving vessel while the steam, being lighter, passes on.

2. Centrifugal-force separators, in which a rotary motion is imparted to the steam, and the water is thus separated from it.

3. Separators having baffle plates, placed in the path of the steam, to which water adheres and from which it then falls to a chamber below the plates.

4. Separators using screens in which the separation is brought about by mechanical filtration.

434. Separator with Baffle. — The baffle type of separator, shown in Fig. 336, consists of an upper and a lower chamber. A single solid baffle is located in the upper chamber, directly across the path of the steam. On the baffle are vertical ribs which prevent the steam current from scrubbing particles of water from the baffle. The steam, after striking the baffle, has its direction changed through 180 degrees and passes through a port at each side of the baffle. The water separated from the steam, runs into the lower chamber and is removed through a drip connection at the bottom.

Receiver Separator. — This type of separator has a large chamber, or receiver, which serves both as a receiving chamber for the water removed from the steam and as a steam reservoir to equalize the steam flow through the steam pipe to the engine. If a receiver separator is installed, a smaller size of steam pipe may be used. The action of this separator usually depends upon the change in direction of steam flow.

The water which collects in separators is generally removed automatically by a float or bucket trap.

Live-steam separators may be located as follows:

1. Inside the boiler, as in marine and locomotive boilers, where the steam piping is short, and rocking of the boiler induces excessive priming. Separators thus used are known as **dry pipes.**
2. Between the boiler and engine, preferably close to the engine. When thus located, a receiver separator is generally used.

435. Oil Separators. — Oil permitted to remain in exhaust steam used for heating accumulates upon the heating surfaces of radiators and lowers their heat-transmitting power. It also forms a condensate unsuitable for boiler feed. Oil separators work upon the same general principle as steam separators, with changes in construction to adapt them to oil. The majority are compact and operate by splitting the steam into many small streams with frequently changing direction. The oil is trapped by baffle plates, so arranged that once the oil is separated it cannot be picked up again by the steam. The oil collects in a reservoir at the bottom of the separator and is removed as in the steam separator.

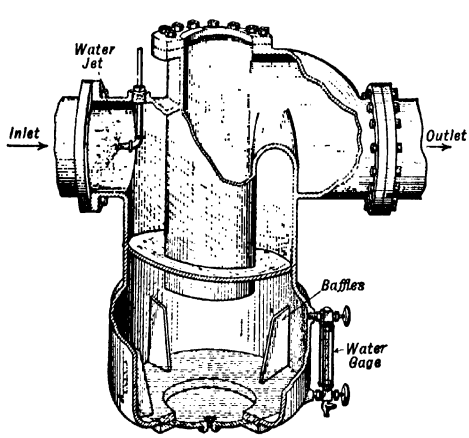

FIG. 337. — Stratton Oil Separator.

The **combined reverse-flow and centrifugal oil separator** is illustrated by the Stratton separator, Fig. 337. It consists of a vertical cast-iron cylinder with an internal central pipe extending from the outlet downward to about half the length of the separator. An annular space is thus formed between the cylinder and the central pipe. The current of entering steam is deflected by a curved partition and thrown tangentially to the side of the annular space. The velocity of the entering steam produces the centrifugal

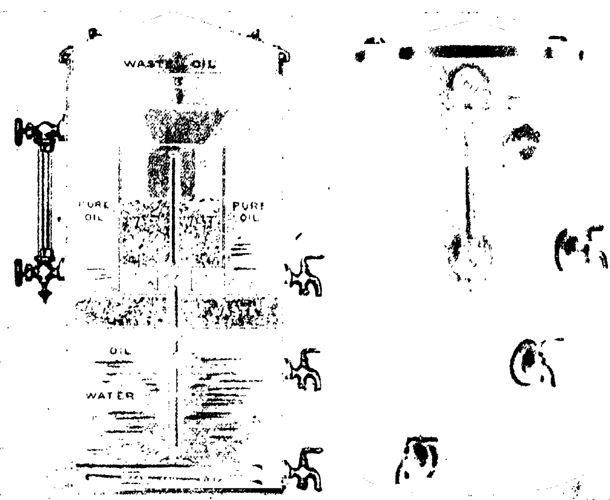

FIG. 338. — Oil Filter.

action, which throws the oil and water to the side, whence it runs down around the outer shell into the lower part of the cylinder. The steam follows the spiral course to the bottom o the central pipe and has its direction of flow changed through 180 degrees. Wings are placed in the bottom of the separator to break up the whirling motion of the water and allow it to settle quietly at the bottom. This separator uses a spray of water to condense any oil in the form of vapor before it enters the separator.

436. Oil Filter. — Oil which drains from bearings contains fine particles which should be removed before the oil is used again. For this purpose an oil filter, of which Fig. 338 is typical, is used. It consists of a cylindrical tank within which are several compartments, one for pure oil, one for waste oil, and one for water. At the bottom of the water chamber is a coil to warm the oil when cold and thus hasten the action of the filter. Waste oil is poured in at the top and is passed through a thick layer of **waste** and into a pipe which leads to the bottom of the water chamber. The oil leaving this pipe is forced to spread out in a thin layer by two **baffle plates,** and is thus

exposed to the action of the water which washes it out. The oil then rises through a second layer of waste into the pure oil chamber near the top of the filter and is ready to be used again.

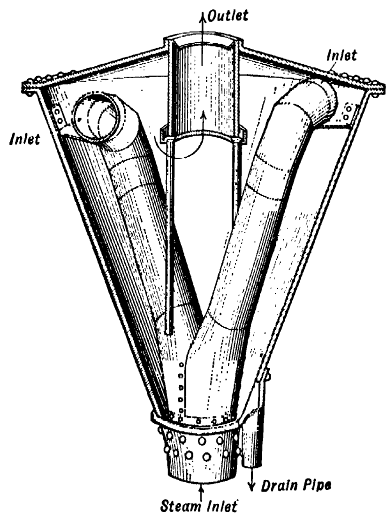

FIG. 339. — Centrifugal Type Exhaust Head.

437. Exhaust Heads. — When engines are operated non-condensing, the oil and water in the exhaust steam is removed by an exhaust head, before the steam is discharged into the atmosphere where the oil is likely to damage the exposed roofs and walls.

Exhaust heads are made of cast iron or sheet steel, the former being generally preferable. Figure 339 shows an exhaust head using the principle of centrifugal force and change in direction of the steam flow. The separated oil and water are removed by a drain pipe connected at the bottom of the chamber.

438. Back-pressure Valve. — This valve prevents excessive back pressure in the exhaust piping from engines, and is called a back-pressure valve

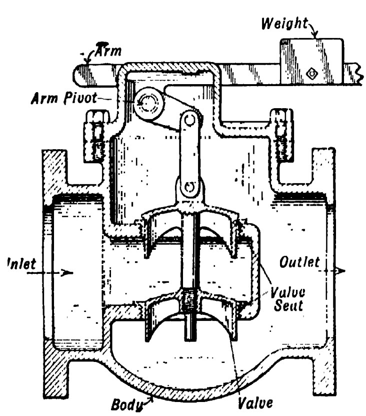

FIG. 340. — Pratt & Cady Lever and Weight Back Pressure Valve

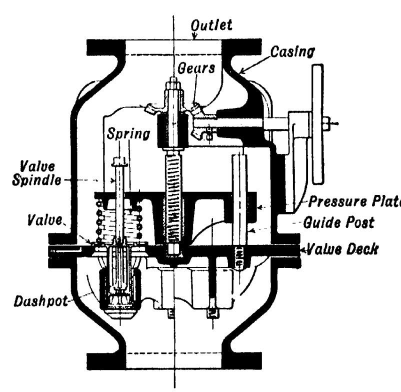

FIG. 341. — Cochrane Multiport Back Pressure Valve.

in non-condensing plants and an **atmospheric relief valve** in condensing plants. There are many types of back-pressure valves, two of which are shown in Figs. 340 and 341. The Cochrane valve consists of several

disks located on a deck plate which carries guide posts for the pressure plate. A dashpot is attached to the underside of the valve to prevent sudden closing. When the pressure in the exhaust piping becomes greater than the pressure of the atmosphere plus that produced by the springs, the valve disks rise and relieve the pressure. The opening pressure can be changed by changing the position of the pressure plate.

The Pratt and Cady back-pressure valve has a weighted lever to regulate the opening pressure. The valve is of the double-disk type having wings to guide it properly. The disks are often provided with a dash pot to prevent pounding.

439. Compound and Vacuum Steam Gages. — These gages, Fig. 342, are used to indicate the pressure of the exhaust steam. The vacuum

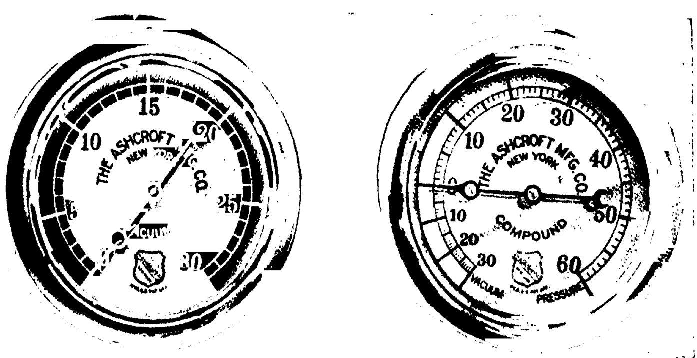

FIG. 342. — Vacuum and Compound Gages.

gage, of either the Bourdon tube or diaphragm type, is used for pressures below that of the atmosphere and is usually calibrated to read in inches of mercury. The construction of a Bourdon type of vacuum gage is similar to that of the pressure gage, but the position of the tube is reversed to make the gage pointer move clockwise. The area of the tube section decreases as the vacuum increases, instead of increasing from its condition at atmospheric pressure, as in the pressure gage.

The compound gage is used to indicate either a pressure in pounds per square inch or a vacuum in inches of mercury. The zero reading on the dial is at the top or at the left side, as in the figure. The needle pinion is under the control of a system of levers so arranged that when under vacuum the needle moves counter-clockwise, and when under pressure, clockwise As generally calibrated, it reads pressure from 0 to 30 pounds per square inch or vacuum from 0 to 30 inches of mercury.

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Lubrication Engineers' Handbook, BRATTLE.
Lubrication, VACUUM OIL CO.
Stationary Steam Engines, VACUUM OIL CO.
Steam Engines, SHEALY.

REVIEW QUESTIONS

1. What is the effect of friction in the moving parts of a steam engine, and how is this effect partially overcome?

2. Name three characteristics of a good lubricating oil.

3. What tests are made upon oils, and what does each tell regarding the lubricating properties of the oil?

4. Name four methods of lubricating engine bearings, and state which is the best and why?

5. Explain the operation of a hydrostatic lubricator. Why is such a lubricator used?

6. Describe the non-pressure oil-circulating system.

7. Name three engine accessories and state the use made of each.

8. Describe the operation of (*a*) a baffle steam separator, (*b*) centrifugal oil separator.

9. What is the purpose of an exhaust head?

10. How is oil prevented from being thrown on the walls and floor of the building by the revolving crank pin of an engine?

CHAPTER XXII

STEAM ENGINE TESTING

440. Foreword. — Tests of steam engines are usually made for the following purposes:

1. To determine the weight of steam used by the engine per indicated horsepower per hour at various loads.
2. To determine the mechanical efficiency.
3. To observe the operation of the engine under different running conditions.

The data obtained from these tests may be used for the same general purposes as those mentioned under Art. 206, page 193.

Apparatus and Instruments Necessary for a Test. — The customary tests made on a steam engine require the following instruments and apparatus to obtain the required observations:

1. Tanks and platform scales to weigh the water, or a suitable water-meter calibrated in place.
2. Graduated scales attached to the water glasses on the boilers, if the steam used is determined by measuring the boiler feedwater.
3. Condenser, to condense the steam in order to obtain its weight, or, in special cases, a steam-flow meter calibrated in position.
4. Thermometers, pressure and vacuum gages.
5. Steam calorimeters.
6. Barometer.
7. Steam-engine indicators.
8. Planimeters.
9. Tachometer, revolution-counter, or other speed-measuring device.
10. Friction brake, or dynamometer if available.

441. Water Measurement. — In making engine tests, it is often desired to measure the cooling water passing through the condenser. *With small condensers, this can be done by weighing; where the quantity of water flowing is large, some form of weir, orifice, nozzle, Venturi tube, or water-meter must be used.* These devices should be carefully calibrated in position.

A satisfactory type of **weir** is shown in Fig. 343. It consists of a 90-degree sharp-edged V-notch, cut in a bronze plate and so placed in a tank that the water to be measured passes over it. The amount of water

passing can be computed by empirical formulae, such as Professor Thompson's formula, which is in general use,

$$Q = 2.544 \sqrt{H^5} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (96)$$

in which

Q = volume of water flowing, cubic feet per second.
H = head on the weir in feet of water.

The **head** on the weir is the height shown in the drawing between the bottom of the V and the surface of the water. This height is obtained by

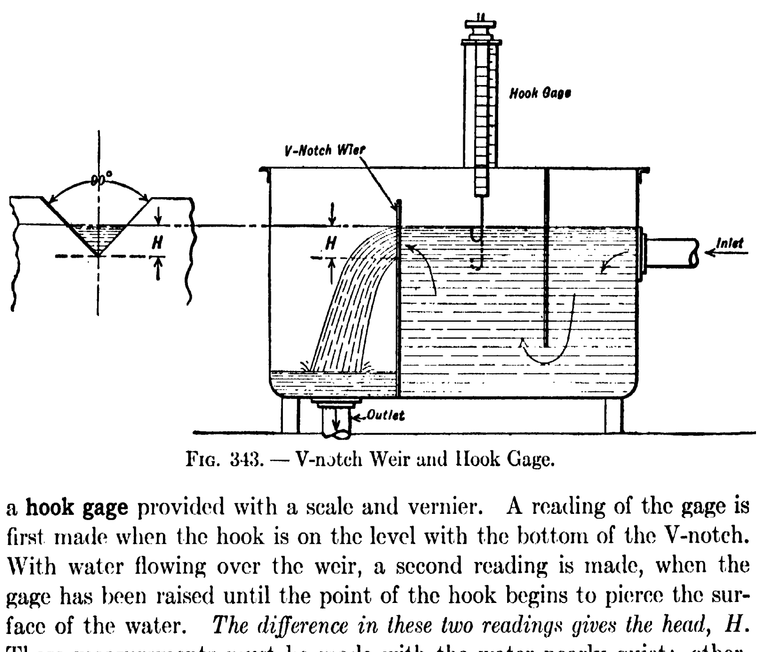


FIG. 343. — V-notch Weir and Hook Gage.

a **hook gage** provided with a scale and vernier. A reading of the gage is first made when the hook is on the level with the bottom of the V-notch. With water flowing over the weir, a second reading is made, when the gage has been raised until the point of the hook begins to pierce the surface of the water. *The difference in these two readings gives the head, H.* These measurements must be made with the water nearly quiet; otherwise a correction must be made for the velocity at which the water approaches the weir.

A typical water-meter, used to measure water under pressure, is shown in Fig. 344. It is known as the **pulsating-diaphragm meter** and consists of an inclined shaft attached to the **diaphragm.** This shaft travels around in contact with a small peg on a plate, which moves the counting mechanism through a system of gears. The frame has side chambers which are alternately filled and emptied, thus producing the " pulsating " motion that operates the recording mechanism. The diaphragm divides the measuring chamber into two compartments of equal volume, one of which is being filled while the other is being emptied. *Meters are generally calibrated to read in cubic feet of water flowing.*

442. Measurement of the Quantity of Steam. — The best method of measuring the steam used is by discharging the steam into a surface condenser and weighing the condensate. When this method is used, all leaks in the condenser should be repaired, or proper correction should be made for the amount of leakage. When a surface condenser is not available, the amount of steam used can be obtained by measuring the feed-water to the boiler supplying the engine In this case, assurance must be had that all the steam from the boiler passes to the engine, and allowance must be made for all water wasted through steam traps or separators located in the steam pipe supplying the engine.

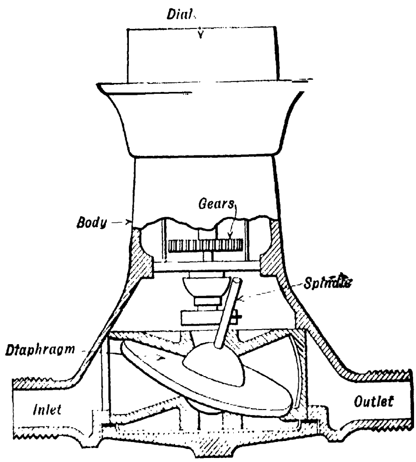


FIG. 344. — Pulsating Diaphragm Water Meter.

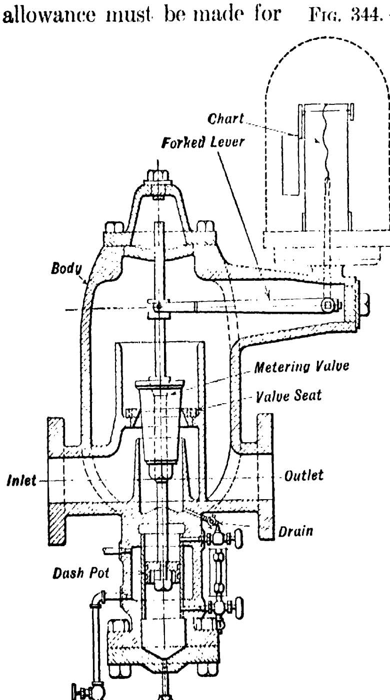


FIG. 345. — St. John's Steam Flow Meter.

Where it is not possible to measure the amount of steam by either of the previously mentioned methods, a **steam-flow meter** connected into the steam pipe line may be used. This instrument is ordinarily calibrated *to indicate the amount of steam flowing in pounds per hour or in boiler horsepower.*

The **St. John's steam-flow meter,** shown in Fig. 345, consists of a cast-iron body connected directly into the steam line. This casting contains the inlet and outlet passages, and has a brass valve seat located in the inlet passage. The valve is a tapered plug attached to a guided valve spindle, which

has a dash pot attached to its lower end to steady the movement of the valve. A forked lever above the valve transfers its movement to a pencil arm, and a clock mechanism moves a calibrated ribbon of paper past the pencil, and makes a record of the amount of steam flow.

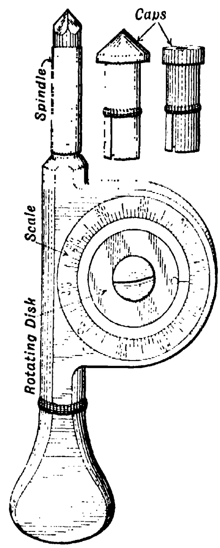


FIG. 346. — Hand Revolution Counter.

This instrument operates on the principle that, with a uniform difference of pressure on two sides of an orifice through which steam is passing with a constant initial pressure, the quantity of steam passing bears a direct relation to the area of the orifice. The valve controlling the opening of the orifice is so tapered that the amount of steam flowing through the orifice, formed by the valve and its seat, bears a direct proportion to the rise of the valve.

Steam enters below the valve and passes through the orifice to the outlet. The current of steam is prevented from impinging directly on the valve, by a deflector cast as a part of the body casting.

443. Methods of Measuring Speed.—The **hand revolution counter,** Fig. 346, is much used for low speeds. It consists of a frame in which a spindle revolves. One end of the spindle carries a worm which meshes with a worm wheel driving a plate calibrated in revolutions; the other end carries a suitable point for

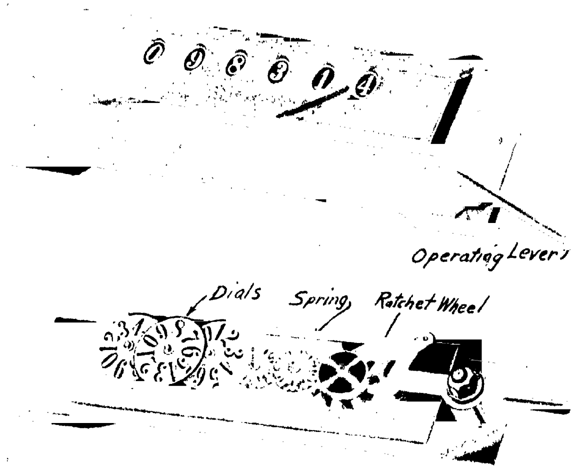


FIG. 347. — Stroke Counter.

connection to the shaft. The point is pressed against the shaft, and the revolutions are counted for one or two minutes. A **stop watch** should be used to observe the time.

The **stroke-counter, Fig. 347,** is suitable for low speeds. It has a frame

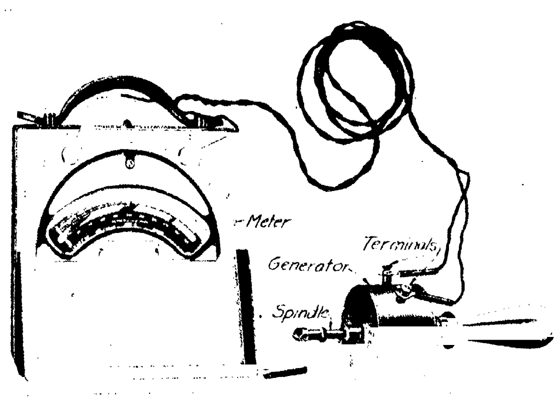


FIG. 348. — Electric Tachometer.

on which are located a train of gears having dials with numbers attached to the gear spindles. The first gear is moved by a **ratchet wheel,** which in turn is moved by a lever with a fixed eccentric pin requiring a certain length of stroke. The outer end of the lever is attached to some reciprocating part of the engine and moves the ratchet wheel one tooth for each revolution. A **spring-pressed pawl** is provided, to prevent the ratchet wheel from moving backward and to hold it in the proper position to permit reading the numbers The first disk reads units, the second tens, the third hundreds, and so on.

For high speeds, an **electric tachometer,** Fig. 348, a portable, geared hand tachometer, Fig. 349, or a chronometric tachometer is used.

The electric tachometer has a permanent magnet, between the poles of which an **armature** revolves. An electromotive force, proportional to the speed, is set up, and is read by a voltmeter calibrated to read revolutions per minute.

The portable instrument shown in Fig. 349, has a case in which are located the operating parts. The main spindle carries a set of gears which can be moved along the spindle by a thumb screw located on the front side of the instrument. Another shaft supports a second set of gears which drive the control portion of the instrument and thus move the indicating hand. The instrument can be set for different speed ranges, by moving the sliding gears to mesh with the various-sized gears on the

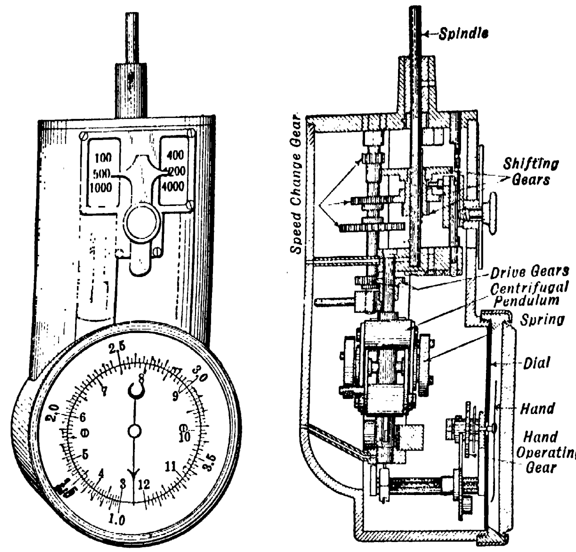


FIG. 349. — Centrifugal Geared Hand Tachometer.

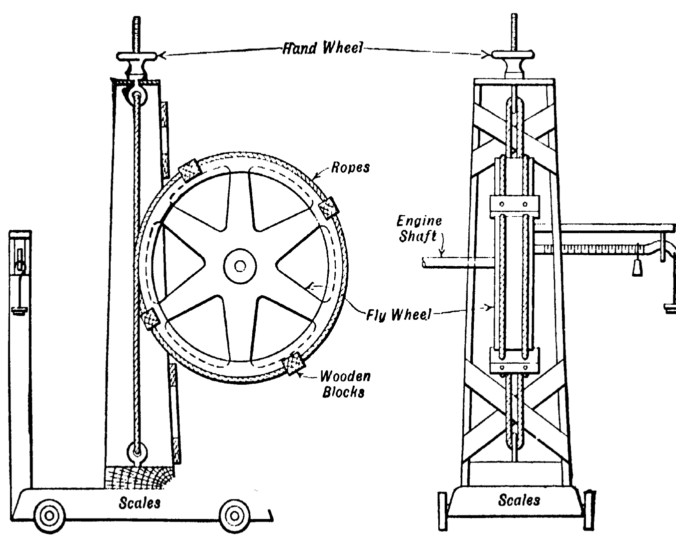


FIG. 350. — Rope Brake

stationary shaft. The operation of this instrument depends upon the change in centrifugal force, acting on the internal rotating parts, at different speeds.

The Von Sicklin-Elgin chronometric tachometer consists essentially of a set of gears, which move the indicating hand over a scale, and an accurate

watch mechanism, which measures the period of time through which the hand moves.

444. Power-measuring Dynamometers. — Besides the brake mentioned in Chapter XIX, a **rope friction brake,** Fig. 350, is often used to measure

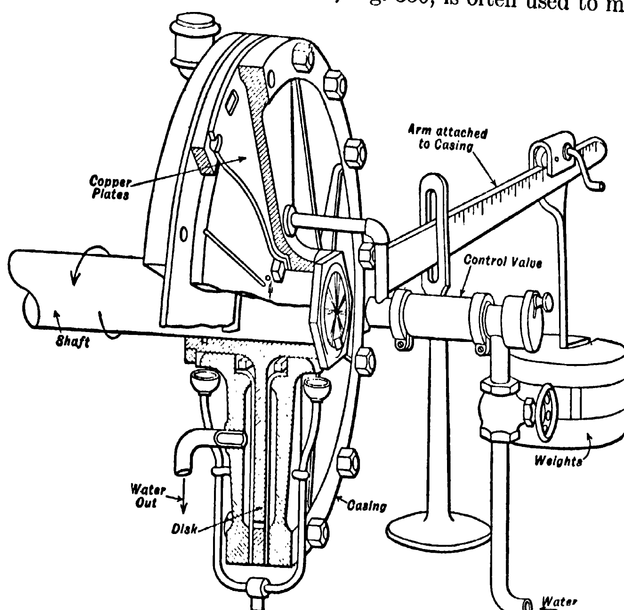


FIG. 351. — The Alden Water Brake.

small amounts of power. The method of using it is evident from a study of the figure.

Another type of brake, or dynamometer, used to measure the power of large engines, is the **water brake,** Fig. 351. This brake has a disk attached to the shaft, transmitting power and revolving in oil between two copper plates. A casting which is free to rotate surrounds the disk and plates, and forms a chamber into which water under pressure is admitted. The pressure of the water forces the plates against the revolving disk, and the water carries away the heat resulting from friction. The friction produced tends to rotate the frame, but is prevented from doing so by weights hung on a calibrated arm attached to the frame. A special valve is provided to maintain a constant pressure of water on the brake, regardless of small pressure changes in the supply. Power is computed in the same way as for the Prony brake.

445. Preparation for the Test. — A careful examination should be made of the engine and the apparatus required in the test. The condition of the inside of the cylinder, valves, and valve seats should be ascertained,

and notes made of all points which might affect the results. The cylinder and valves should be tested for leaks, and if a condenser is used its packing should be examined for air leaks. For this purpose a candle flame may be used to detect the leaks, or the piping and condenser may be filled with warm water under sufficient pressure to show the leaks if any exist.

The diameters of the engine cylinders should be obtained when cold, and the clearance obtained by finding the weight of water required to fill the clearance space. To determine the clearance by this method, the engine is set on dead center, and the clearance space filled from a quantity of water previously weighed. The water remaining when the clearance space is filled is subtracted from the original weight, to give the weight of water required. In converting this weight to volume, allowance must be made for the temperature of the water used. All air pockets should be vented to obtain accurate results. If there is leakage past the piston and valves, its amount should be allowed for by having the clearance space full and allowing it to settle for a certain length of time, then filling the clearance space again and obtaining the amount used to refill. The time required for filling, leakage, and refilling should be taken. The amount of water, W_1, required to fill the clearance space is then

$$W_1 = W - \frac{wT}{t - t_1} \qquad (97)$$

in which

W = weight of water poured into clearance space.
w = weight of water to refill.
T = time to fill clearance space.
t = time for leakage.
t_1 = time to refill.

For large engines it is necessary to compute the clearance volume from working drawings of the engine.

If a surface condenser is used to determine the amount of steam used, it should be examined for leakage in the condenser tubes, by operating the condenser under vacuum with all steam shut off and observing the rate at which water is discharged by the condensate pump.

All instruments should be carefully calibrated before and after test.

The indicator springs should be calibrated under conditions as similar as possible to those prevailing in actual use. They should preferably be compared with a standard dead-weight tester, or a standard gage known to be correct. The calibration should be made for at least five equidistant points, and the arithmetical mean should be used as the average scale.

Platform scales should be compared with standard weights.

446. Conduct of Test. — The conditions under which the test is to be run should be maintained throughout.

The test should not be started until all the apparatus has been in operation for a sufficient length of time to be thoroughly warmed, and uniform conditions have been established. The time should then be noted and the observations started. The test should continue from three to ten hours, depending on conditions, and readings should be taken at least every fifteen minutes, or oftener if conditions fluctuate. Wide fluctuations should be recorded by recording instruments.

Each indicator card should be marked with the date, make and size of engine, number of card, time, scale of spring, and end of cylinder. One card of each set should be marked with the readings of the steam and vacuum gages.

A log of the test should be made, on which should be entered readings of steam and vacuum gages, thermometers, calorimeters, speed indicators, load-measuring devices, and steam used. These readings should be obtained at practically the same time the indicator diagrams are taken. The areas, lengths, mean effective pressures, and cut-offs shown by the diagrams should be placed on the log, and a set of representative indicator diagrams should be selected for inclusion in the record.

447. Calculation of Results. — *Before any calculations are made, the data should be examined carefully, and any observations which are obviously incorrect should be thrown out.* In making calculations involving the averages of a number of readings, the following methods may be used:

1. Primary averages in which all the readings of each instrument are averaged and the resulting averages used to calculate the final results.
2. Final averages in which the final result is calculated from each set of coincident observations. These final results are averaged as a grand average.

The choice between these two methods depends upon the degree of accuracy desired, and the type of formula involved. With a formula involving the sum or difference of first powers, either method is satisfactory; when the fo mula involves fractional powers or powers greater than the first, the method o final averages should be used to obtain the correct result.

Each item is calculated according to the methods explained in the previous chapters. The observations and the results should be tabulated in a form, such as shown in Table 32, page 449, in which roman figures indicate observed values and **bold-faced** type shows computed values.

To determine the point of cut-off from the diagram, the method shown in Fig. 352 is used. Through the point of maximum pressure during admission, a line is drawn parallel to the atmospheric line. Through a point on the expansion line where cut-off is complete, a **hyperbolic curve,**

Art. 396, page 397, is drawn. The point at which the line through the point of maximum pressure intersects the hyperbolic curve is the point of **nominal cut-off.** The proportion of cut-off is found by dividing the length up to the point of cut-off, l, by the total length, L.

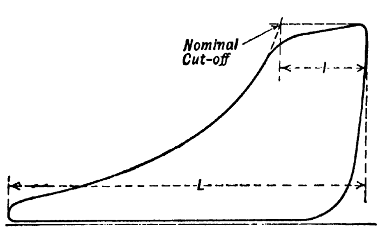


FIG. 352. — Method of Finding Point of Cut-off.

448. Calculation of a Heat Balance. — For some purposes it is often desired to make a heat balance for the engine, using the data obtained during the test. Such a heat balance is shown in Table 31, and the method of calculating the various items is illustrated. The data used in this heat balance were taken from the form given in Table 32.

TABLE 31. — HEAT BALANCE FOR AN 8 IN. BY 18 IN. CORLISS ENGINE

Name of Loss	B.t.u.	Per Cent
Heat supplied to engine per hour	915,018	100
Heat carried away by condensing water per hour	814,312	88.9
Heat equivalent of the i.hp. per hour	67,241	7.4
Heat lost by radiation, etc.	33,465	3.7

Heat supplied to the engine per hour equals the weight of steam used by the engine per hour multiplied by the quantity of heat per pound of steam above 32 deg. fahr. at the absolute pressure and quality in the steam main.

Example 48. — Weight of steam used, 780 pounds per hour; quality equals 0.977; absolute steam pressure, 134.5 lb. per sq. in.; exhaust pressure, 15.6 lb. per sq. in. abs. Find the heat supplied per hour.

Solution. — Heat per pound $= x L_1 + h_1 = 0.977 \times 872.3 + 320.9 = 1173.1$ B.t.u. Heat supplied, B.t.u. per hour, equals $780 \times 1173.1 = 915{,}018$.

Heat absorbed by the circulating water per hour equals the weight of circulating water multiplied by the rise in temperature of the water in passing through the condenser. As an equation,

$$\text{Heat absorbed} \doteq W_c (t_2 - t_1) \quad \ldots \ldots \quad (98)$$

in which W_c = weight of circulating water per hour, pounds.
t_1 = temperature at outlet, deg. fahr.
t_2 = temperature at inlet, deg. fahr.

Example 49. — W_c as observed during test given in Table 33, on page 449, = 21,715 lb. per hr. t_1 = 55.5 deg. fahr.; t_2 = 93.0 deg. fahr.

Find the heat absorbed by the circulating water per hour.

Solution. — Heat absorbed = 21,715 (93.0 − 55.5) = 814,312 B.t.u.

Heat equivalent of the i.hp. per hour equals the number of indicated horsepower multiplied by the heat equivalent of one horsepower hour

$$= \text{i.hp.} \times \frac{33{,}000 \times 60}{777.5} = \text{i.hp. per hour} \times 2547$$

Example 50. — The indicated horsepower, as shown by the engine test, Table 32, equals 26.4. Find the heat equivalent of the i.hp.

Solution. — Heat in B.t.u. = 26.4 × 2547 = 67,241.

Heat lost by radiation and other causes equals the heat supplied per hour minus the sum of the heat absorbed by the condenser per hour and the heat equivalent of the i.hp. per hour.

Example 51. — Using the data in the three previous examples, find the heat lost by radiation per hour.

Solution. — Heat supplied per hour, from Example 49 = 915,018. Sum of items in Examples 49 and 50 = 814,312 + 67,241 = 881,553. Heat lost by radiation, etc., including friction = 915,018 − 881,553 = 33,465 B.t.u.

TABLE 32. — DATA AND RESULTS OF RECIPROCATING STEAM ENGINE TEST

A. S. M. E. Test Code, 1920

Item	Name of Item with Units	
	GENERAL INFORMATION, DESCRIPTION AND DIMENSIONS	
1	Date of test	Mar. 10, 1919
2	Builder of engine	Murray Iron Works
3	Object of test	Performance
4	Type of engine	Simple
5	Class of service	Mill
6	Auxiliaries	Steam driven
7	Type and make of condenser equipment	Wheeler, surface
8	Rated capacity of condenser	....
9	Rated power of engine	25
10	Kind of valves	Corliss
11	Type of governor	Pendulum
12	Diameter of cylinder, in.	8
13	Diameter of piston rod, in.	$1\frac{7}{8}$
14	Stroke of piston, in.	18
15	Clearance volume of head end, per cent	5.56
16	Clearance volume of crank end, per cent	5.10
17	Horsepower constant, head end	**0.00228**
18	Horsepower constant, crank end	**0.00216**
	TEST DATA AND RESULTS	
19	Duration of test, hr.	3
	Average pressures	
20	Barometric pressure, in. of mercury	29.51
21	Barometric pressure, lb. per sq. in	**14.4**
22	Pressure in steam pipe near throttle, lb. per sq. in., gage	120
23	Absolute pressure corresponding to item 22, lb per sq. in.	**134.4**
24	Pressure in exhaust pipe near engine, lb. per sq. in.	0.10

TABLE 32. — DATA AND RESULTS OF RECIPROCATING STEAM ENGINE TEST *Continued*

A. S. M. E. Test Code, 1920

Item	Name of Item with Units	
25	Absolute pressure corresponding to item 24, lb. per sq. in.	**14.5**
	Average temperatures	
26	Engine room temperature, deg. fahr.	73
27	Temperature of saturated steam corresponding to pressure in exhaust pipe near engine, deg. fahr.	211.6
	Quality of Steam at throttle	
28	Per cent of moisture in steam, per cent	2.30
29	Quality	**0.977**
	Total quantities	
30	Total steam consumed by engine, as measured, lb.	2340
31	Total dry and saturated steam consumed, lb.	**2286**
	Hourly quantities	
32	Steam consumed per hour as measured, lb.	**780**
33	Dry and saturated steam consumed per hour, lb.	**762**
	Heat consumption	
34	Total heat above water at 32 deg. fahr. per pound of steam at throttle, B.t.u.	**1173.1**
35	Heat of liquid at temperature of steam at exhaust pressure, B.t.u.	**180.0**
36	Heat supplied per pound of steam, B t.u.	**1173.1**
37	Heat consumed per hour, B.t.u.	**774,600**
38	Heat available for work per lb. of steam from adiabatic expansion between initial conditions and final pressure according to the Rankine cycle, B.t.u.	**163.3**
	Indicator diagrams	
39	Nominal cut-off, per cent	**19.6**
40	Mean effective pressure (average), lb. per sq. in.	
	Speed	
41	Revolutions per minute, r.p.m.	102
	Power	
42	Indicated horsepower developed by whole engine, i.hp.	**26.4**
43	Brake horsepower developed by whole engine, b.hp.	**25.6**
44	Friction of engine, hp.	**0.80**
45	Mechanical efficiency, per cent	**97**
	Economy results	
46	Steam consumed per i.hp.-hr. as measured, lb.	**29.5**
47	Dry and saturated steam consumed per i.hp -hr., lb.	**28.9**
48	Steam consumed per b.hp.-hr. as measured, lb.	**30.4**
49	Dry and saturated steam consumed per b.hp.-hr., lb.	**29.8**
50	Heat consumed per i.hp.-hr.	**29,300**
51	Heat available according to Rankine cycle per i.hp.-hr., B.t.u.	**4820**
	Efficiency results	
52	Thermal efficiency referred to i.hp., per cent	**8.7**
53	Engine efficiency based on i.hp., per cent	**53.3**
54	Rankine cycle efficiency, per cent	**16.2**

REFERENCES

Power Test Code, A. S. M. E.
Power Plant Testing, MOYER.
Mechanical Laboratory Methods of Testing, SMALLWOOD.
Experimental Engineering, and Manual for Testing, CARPENTER AND DIEDERICHS.

REVIEW QUESTIONS AND PROBLEMS

1. Name the apparatus required to make the observations necessary for a performance test of a steam engine.

2. State the purpose of each instrument named in Question 1.

3. Name three methods used to measure water, and describe one of them.

4. In testing a centrifugal pump, the water discharged was measured by a 90-deg. V-notch weir. The hook gage reading was 1.762 in. at maximum discharge, zero reading 1.4537 in. Using Professor Thompson's formula, compute the flow in pounds of water, the observed temperature of which was 67 deg. fahr.

5. Describe one form of steam meter. Under what circumstances should it be used to obtain the weight of steam used by an engine?

6. Check items given in **bold-faced** type in Table 33.

7. Explain the meaning of items 39, 46, 52 and 53.

8. What is meant by "the method of averages," as applied to computing the results of a test.

9. Mention four methods of measuring speed. Which method should be used to give accurate readings at high speeds?

10. Explain the water method of finding the clearance of an engine.

CHAPTER XXIII

STEAM TURBINES

449. Foreword. — A steam turbine is a steam engine having a rotating wheel or cylinder to which is fastened a series of buckets,* uniformly spaced, on its periphery. Steam from nozzles or guide passages is directed continuously against these buckets, thus causing rotation. Expansion of the steam in the nozzles or buckets converts its heat energy into energy of motion, and gives it a high velocity, which is expended on the moving blades. *The difference in the various types of turbines is due to different methods of using the steam, depending upon the construction and arrangement of the nozzles, steam passages, and blading.*

The steam turbine is essentially a high-speed machine It is used to best advantage with direct connection to electric generators, centrifugal pumps and compressors; and with geared connections to marine propellers, rolling mills and machinery which should run at low speed.

Steam turbines range in capacity and speed from a few horsepower at 20,000 or 30,000 revolutions pe minute, in the smaller, single-wheel turbines, to the modern three-cylinder cross-compound turbine having a continuous rating of 60,000 kw., with the rotors in the separate cylinders running at 1800 and 1200 revolutions per minute.

The advantages claimed for the steam turbine are comparatively low initial cost, low expense for maintenance and attendance, small floor-space requirement, light foundations, large overload capacity, freedom from oil in the exhaust steam, absence of vibration, uniform velocity of rotation, and high efficiency over a wide range of load in large installations. The steam turbine can be built in units of greater capacity than is practical with reciprocating steam engines.

450 Classification of Steam Turbines. — Turbines may be classified in the following ways:

1. By position of shaft	Horizontal Vertical
2. By pressure of the entering steam	High pressure Low pressure

* The terms **blade, bucket** and **vane** are used interchangeably by various manufacturers of steam turbines. Blade is generally applied to reaction turbines, and bucket to impulse turbines. The term bucket will be used in this chapter to avoid confusion.

3. In accordance with the action of the steam	Impulse Reaction Combined impulse and reaction
4. By method of subdividing the flow of energy in the buckets	Pressure stage Velocity stage
5. By direction of flow of the steam with relation to the bucket wheel	Axial flow Single flow Double flow Tangential flow Radial flow
6. By number of cylinders used to expand the steam	Single cylinder Compound cylinder Multiple cylinder

451. General Types of Steam Turbines. — In an **impulse type of turbine** *the expansion and consequent change in pressure of the steam occurs entirely within the nozzles, which direct the steam in jets against the moving buckets.*

In a **reaction type of turbine** *the steam is directed against the moving buckets by guide vanes or orifices, and the pressure of the steam changes as the steam expands through both stationary guide vanes and moving buckets.*

The forces of impulse and reaction are present in both these types, and the designation is made in accordance with the predominating effect. The meaning of these terms can be made clear by considering a jet of steam issuing from a stationary nozzle and impinging against a movable flat plate. The force causing the plate to move is due to impulse, because it is a force acting in a forward direction. The jet as it leaves the nozzle exerts a reaction, or backward, force upon the nozzle. As applied to commercial turbines, impulse and reaction are illustrated by a nozzle, Fig. 353*b*, discharging against a movable curved plate. The force causing motion of the plate is a combination of impulse and reaction, and is greater than the impulse or the reaction force acting alone. Buckets are thus made with a shape that reverses the jet of steam as much as is practical, thereby increasing the force acting on the bucket. The forms of buckets used in commercial impulse and reaction turbines are shown in Fig. 353*a* and *c*.

In a **multi-pressure stage turbine** the total drop in pressure is divided into a number of small pressure drops, each of which is used as though in a separate turbine, and all the wheels of the turbine are carried on a common shaft. In the impulse type of pressure-stage turbine, the total drop in pressure of the steam, from inlet to exhaust outlet, is divided between two or more sets of impulse-type nozzles, each set having its row

of moving buckets. The reaction turbine, from its construction, is a multi-pressure stage turbine, each stage being considered as made up of a single row of stationary vanes and its corresponding row of moving buckets. The multi-pressure stage construction is also called pressure compounding.

In the **multi-velocity stage turbine,** there are two or more rows of moving buckets to take up the velocity of the jets of steam, which is expanded from initial to final pressure in one set of nozzles. The steam is directed

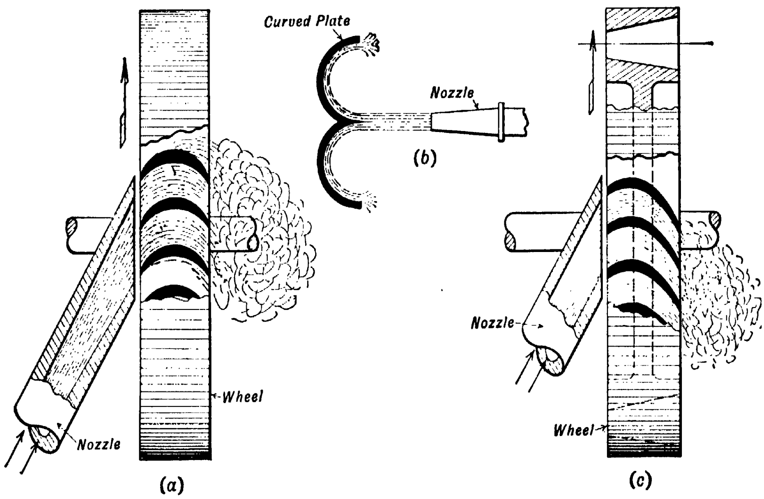


FIG. 353. —Action of Jet in Steam Turbines, and Commercial Forms of Impulse and Reaction Buckets.

by the nozzles upon the first row of moving buckets. The steam leaving the first row of buckets is re-directed against a second row of moving buckets by the first row of stationary vanes, and so on through the various rows of buckets used to absorb the velocity.

The object of multi-staging is to reduce the rotative speed of the turbine shaft. When the entire expansion occurs in a single set of nozzles, as in the De Laval single-stage turbine, using a single row of buckets, the velocity attained by the jet of steam is between 3500 and 4000 feet per second. The velocity of the corresponding buckets is between 1200 and 1400 feet per second for best economy. This speed is prohibitive, except with wheels having a small diameter,* on account of the strength of the materials required. By using multi-stage velocity turbines, the velocity of the buckets may be made less, since the initial velocity is taken up in

* In the De Laval single-stage turbine the diameter of the wheel, to the center of the buckets, in a 5 hp. turbine, is 3.94 inches, and the rotative speed is 30,000 r.p.m.

several rows with a smaller amount per row. This permits using wheels of larger diameter with lower bucket speeds.

The same result is obtained by using pressure staging, since the pres-

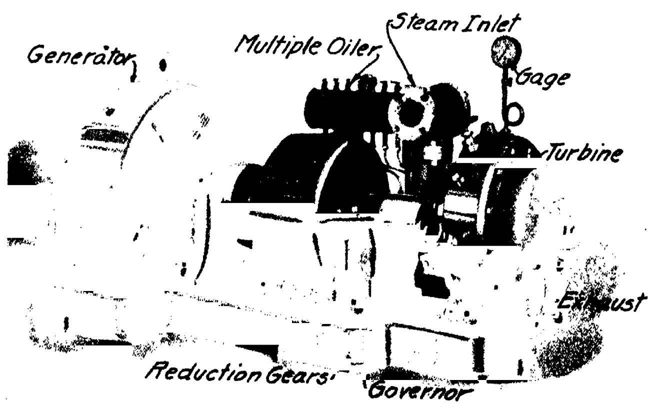


FIG. 354. — 35-kw. De Laval Impulse Steam Turbine with Reduction Gear.

sure drop per stage is only a portion of the total, and the velocity to be taken up by each row of buckets is correspondingly reduced.

452. Simple Impulse Steam Turbine. — A simple form of De Laval

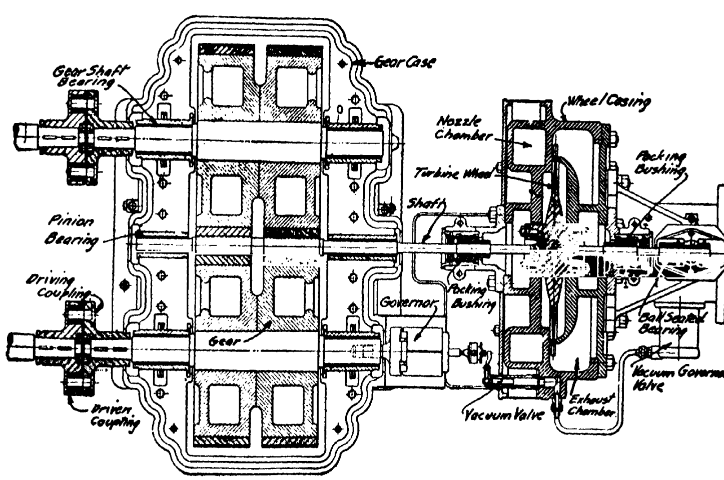


FIG. 355. — Horizontal Section of De Laval Double Geared Turbine.

steam turbine representative of the elementary turbine, is shown in Figs. 354 and 355. It has a steel casing with three bearings supporting a

flexible shaft, which carries a nickel-steel wheel having a single row of drop-forged steel buckets, around it periphery. A section of the rim of the wheel, Fig. 356, shows the buckets with bulb shanks forced into slots in the wheel. The upper ends of the buckets have projections, which fit closely together, forming a continuous ring around the circumference. On account of the high velocity of rotation, 20,000 to 30,000 r.p.m., the shaft is made light and flexible to permit the wheel and shaft to rotate about the center of mass and thus to prevent vibrations.

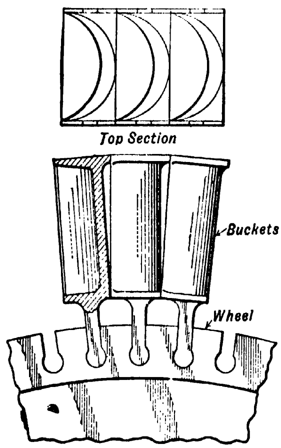


FIG. 356. — Buckets of De Laval Turbine and Method of attaching to Wheel.

The bearings are made in halves, lined with anti-friction metal, reamed inside, and ground true outside. The outer bearing carries most of the weight on the shaft and has a spherical seat in the casing. This gives flexibility and makes the bearing self-aligning. A helical spring, held in position by a cap, presses the middle bearing against its seat. The inner bearing is flexible and free to oscillate, and acts as a stuffing box to prevent leakage of steam from the casing. These bearings are oiled from a central reservoir.

Small disk wheels have holes through the center, and are forced on tapered sleeves shrunk on the shaft. The larger wheels are made solid for greater strength, and are bolted to flanges on the shaft. The thickness of the wheel section is increased from the rim to the center to aid in equalizing the stresses in the wheel. At the base of the rim, below the buckets, it is made thin enough to confine a break to that part of the wheel, in case of damage to the row of buckets.

Steam passing through the **throttling governor,** Fig. 357, enters the **expanding nozzles,** in the steam chest, which are controlled by needle valves adjusted by hand. The number of nozzles depends on the size of the turbine, and ranges from 1 to 15. The cross section of the nozzles is generally round, though square or rectangular sections are sometimes used; their shape is such that the steam attains a maximum velocity during expansion. The nozzle is located to direct the high-velocity jet against the bucket, at the proper angle to give minimum impact at entrance.

The speed of the turbine shaft, which is too high to be properly utilized, is reduced by means of **reduction gearing.** The pinion has **helical teeth** cut on an enlargement of the flexible shaft. It meshes with and drives

a large helical gear on a shaft directly coupled to the generator shaft. The teeth on both pinion and gear are cut in two rows, one row cut right-hand and the other left-hand, at an angle of 45 degrees. This construction is necessary for quiet and efficient operation at high speeds and prevents end thrust, or endwise movement of the shaft.

The gear case is substantial, and carries ample bearings for the pinion and gear shaft. The gear centers are of cast iron with steel rims shrunk

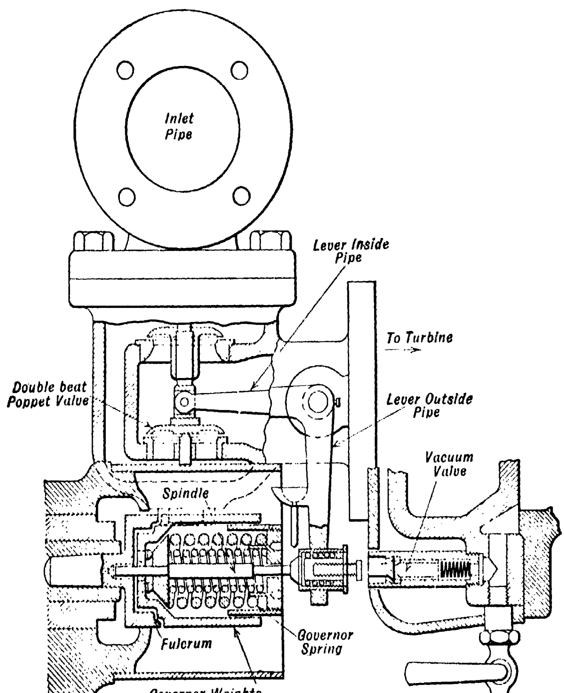


FIG. 357. — Throttle Valve and Governor De Laval 35-kw. Turbine.

on them. The teeth are cut on the rim with great accuracy and, if they are maintained in proper alignment with suitable lubrication, the wear after long periods of operation is slight, since the pressure on the teeth at the high speed of rotation is low.

The **governor** is a simple centrifugal throttling governor, Fig. 357, generally attached to the slow-speed shaft of the reduction gear. *It controls the speed by regulating the pressure of steam admitted to the nozzles.* The governor weights are cylindrical and surround a helical spring. This spring rests on a collar, against which pins in the governor weights

bear. Knife-edges on the governor weights bear against a fulcrum on the main governor support. When the speed increases above normal, centrifugal force causes the governor weights to swing farther apart. This compresses the governor spring and moves the central spindle to the right, thus moving the bell-crank lever and operating the double-disk poppet admission valve to reduce the opening for steam, thus reducing

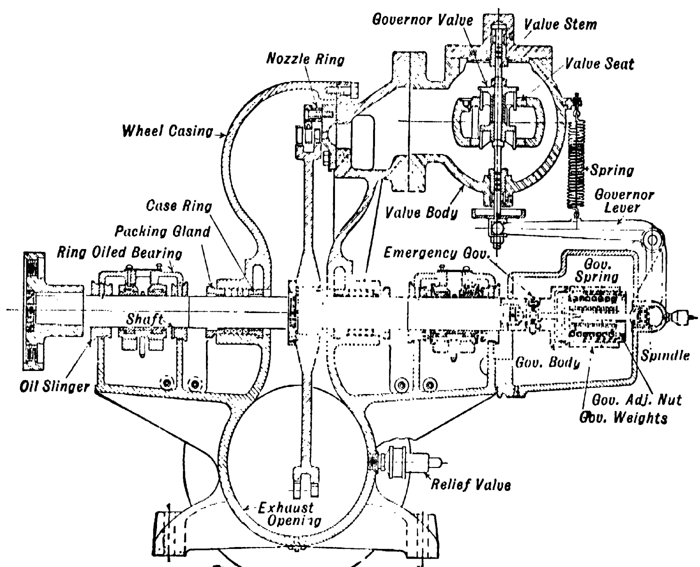


FIG. 358. — De Laval Velocity-stage Impulse Turbine.

the speed. The admission valve is normally held open by a spring attached, outside, to the bell crank. An adjusting nut screwed into the governor body controls the initial compression of the governor spring, and hence the speed of the turbine.

When the turbine is operated condensing, quick closing of the admission valve in case of emergency does not decrease the speed rapidly enough. Therefore, it is arranged so that further movement of the governor pins compresses a stiff spring in the stationary bell crank, and opens a vacuum valve. This admits air to the wheel chamber and decreases the speed by reducing the vacuum. This single-wheel turbine is built in sizes up to 600 horsepower, with peripheral speeds as high as 1300 feet per second.

The more recent types of De Laval turbines are of the velocity or pressure-stage construction and will be described as illustrative of these types.

453. De Laval Velocity-stage Impulse Turbine.—The velocity-stage De Laval turbine, Fig. 358, has either two or three velocity stages, with the

steam wholly expanded in one set of nozzles and the resulting velocity taken up in the rows of moving blades, with the necessary stationary guide vanes for re-directing the steam upon the moving buckets. The forms of the

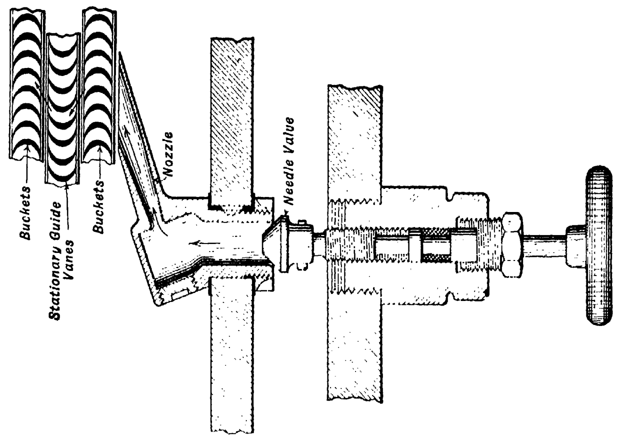


FIG. 359. — Nozzle and Bucket Arrangement of De Laval Velocity-stage Turbine.

nozzles are shown in Fig. 359. The buckets are similar to those described in the previous article. The shaft to which the bucket wheels are keyed

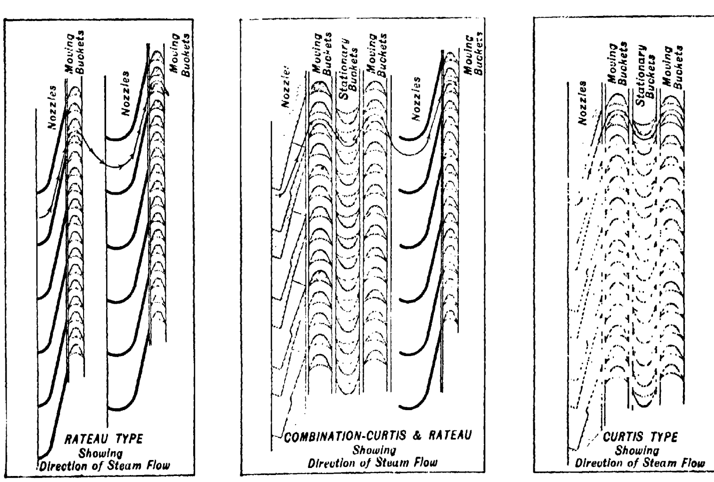


FIG. 360. — Rateau and Curtis Turbine Elements.

is short and stiff, with a diameter sufficient to prevent vibration. It is supported at each end by a plain ring-oiled bearing. Thrust collars at

the forward bearing prevent end play of the shaft. Leakage of steam around the shaft, from or into the casing, is prevented by **metallic ring packing** held in place by a gland. The speed is regulated by a throttle valve under the control of a centrifugal governor mounted directly on the turbine shaft.

454. Multi-pressure Stage Impulse Elements. — The impulse elements, consisting of sets of nozzles and the corresponding buckets, for medium- and large-sized pressure stage turbines, are of either the **Rateau** or **Curtis type,** Fig. 360. In the Rateau type a pressure stage is formed by expanding the steam in a single set of nozzles and taking its velocity up by

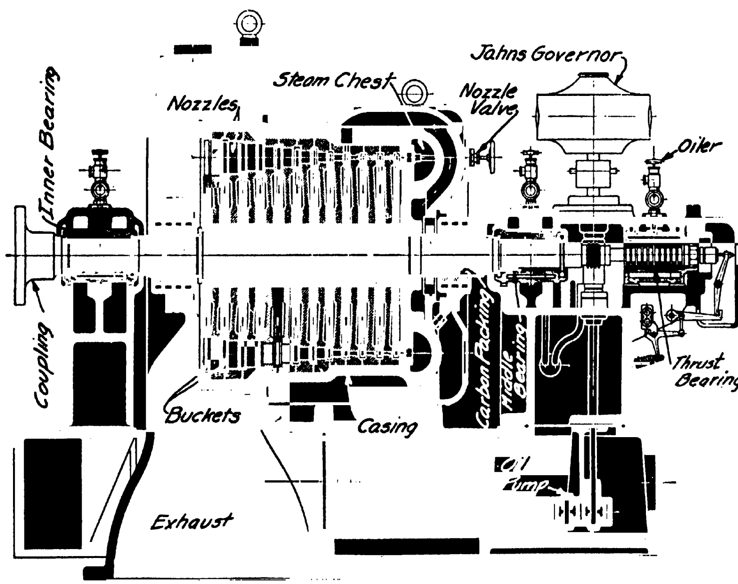


FIG. 361. — De Laval Pressure-stage Impulse Turbine.

a single row of buckets, the number of stages depending upon the capacity desired. In the Curtis type, the steam is expanded in a single row of nozzles to give it sufficient velocity to carry it through two rows of moving buckets and one row of stationary buckets, the nozzles and buckets forming one pressure stage.

Each type has its advantages. The Curtis type is especially suitable for high initial pressure. A combination of these types is used on large Curtis and Kerr turbines.

455. De Laval Pressure-stage Impulse Turbine. — This turbine, which is made in sizes from 50 to 15,000 horsepower, is illustrative of the Rateau type of construction, the general arrangement of the parts being shown in Fig. 361. A heavy shaft carries separate bucket wheels, each

revolving in a separate chamber between diaphragms held by a cylindrical casing. The wheels are mounted on the shaft by means of tapered bushings which are drawn into position by a nut. Three bearings support

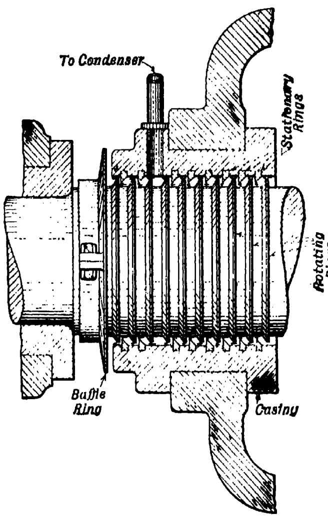


FIG. 362. — Labyrinth Packing.

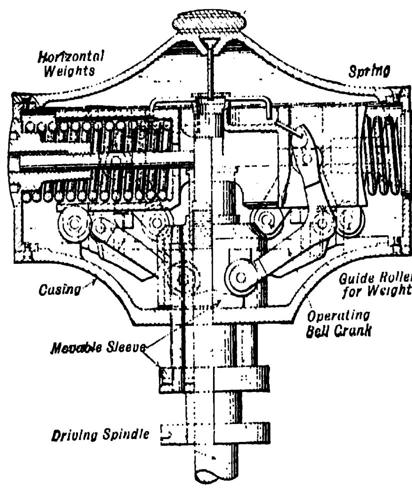


FIG. 363. — Jahns Governor used on De Laval Multi-stage Steam Turbines.

the shaft, two of which are ring-oiled and the third a marine type of thrust bearing. To reduce the leakage of steam between stages and from the casing at the high-pressure end, removable **labyrinth packings,** Fig. 362, carried by the diaphragm and casing, surround the rotating hub and shaft. This type of packing consists of metallic rings carried by the shaft, between which are stationary rings supported by the frame. On account of the small clearances between the rings and shaft and the long zigzag path for escape of steam, the leakage is small. The space between the shaft and rings may be piped to a lower pressure stage or to the condenser. Condensation passing by the packing rings is deflected from the bearings by a **baffle ring.**

The flow of steam to the steam chest is regulated by a double-seated poppet throttle valve under control of a centrifugal governor, Fig. 363, of the Jahns type, mounted on a vertical spindle driven from the main shaft by worm gearing. It consists of cylindrical hollow weights surrounding spiral spr ngs and moving in a horizontal plane. The weights are attached to bell cranks which move a sliding sleeve connected to the throttle valve. Roller bearings guide the bell cranks both horizontally and vertically.

Steam admitted to the steam chest passes through the first-stage

nozzles, which occupy only a part of the circumference and may be controlled by hand-operated valves. After leaving the first row of moving buckets, the movement of the steam is reversed in the **stationary guide vanes,** which are set in around the entire circumference of the **diaphragm.** As it leaves the guide vane, the steam is expanded again in short nozzles formed between the vanes in the diaphragm, and is thus re-directed against

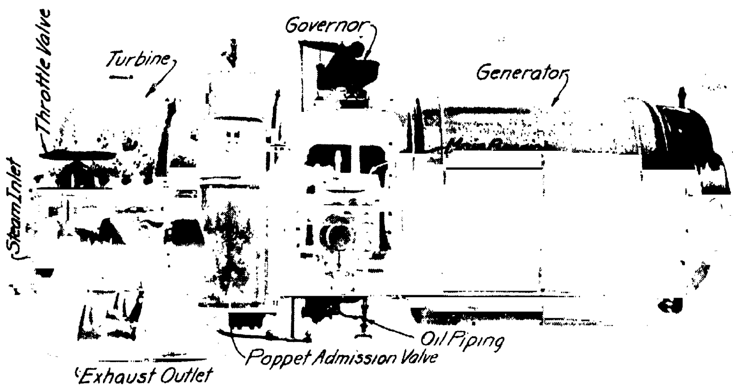


FIG. 364. — Outside View of 3000-kw. Curtis Turbine.

the second row of moving buckets. The path of the steam through the stages, each consisting of a row of nozzles and buckets, is continuous until the pressure has fallen to that of the steam at the exhaust connection. The height of the buckets increases as the exhaust end is approached, to allow for the increase in volume of the steam.

456. Curtis Impulse Turbine. — Curtis turbines are of the pressure-stage impulse type, ranging from the single-stage turbine of small power up to 18 or 20 stages in the larger 45,000-kw. units. The 5000 to 10,000-kw. turbines were formerly made with vertical shafts, but in recent years the horizontal construction has been used. The smaller turbines have the Curtis type of multi-pressure staging, and the larger turbines have a combination of the Curtis and Rateau types.

Outside and sectional views, Figs. 364 and 365, show the general construction of a medium sized horizontal Curtis turbine. The casing enclosing the wheels is made of cast iron, and carries the bearings which support the shaft. The casing is split horizontally, and the upper and lower sections are held together with bolts Non-conducting material, covered by sheet-metal, is placed over a part of the casing. The location of the steam-operated valve gear, governor, governor shaft and steam connections is shown.

The rotating element, consisting of four steel disks or wheels, mounted

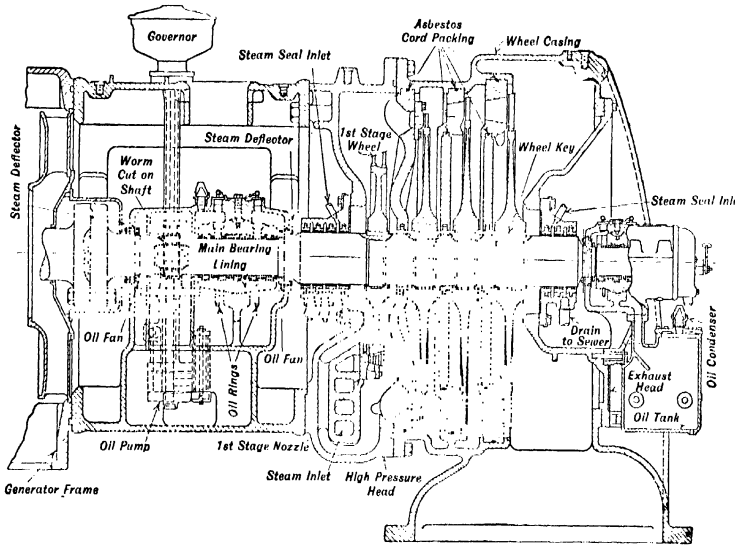


FIG. 365. — 4-Stage Condensing Curtis Steam Turbine — Sectional View.

on a stiff shaft, is shown in Fig. 366. The first wheel carries two rows of buckets, arranged on the wheel as in Fig. 367*a* and *b*; the other wheels carry one row each. Each wheel runs in a separate chamber, formed by diaphragms supported between the wheels by the frame of the turbine.

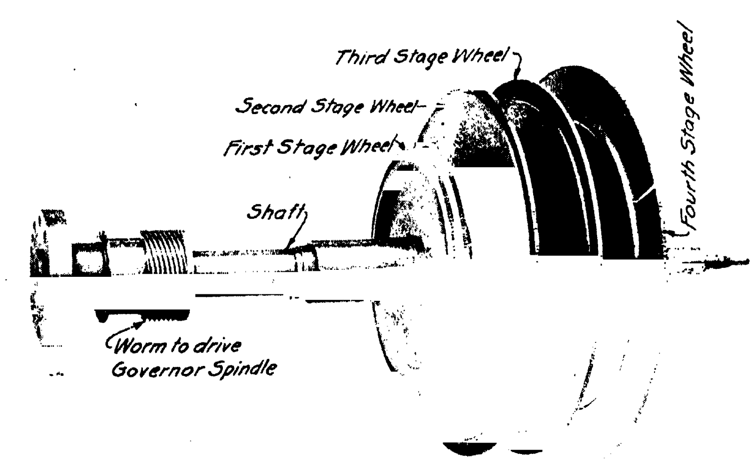


FIG. 366. — Rotating Element of 4-stage Curtis Turbine.

The buckets are made of steel alloyed with nickel, vanadium, or chromium, the composition depending upon the strength required by the bucket sections. They are drop-forged to exact dimensions, and the **shanks** are then milled to form a **dovetail tenon,** which is inserted into a corresponding slot in the rim of the wheel. The tips of the buckets pass through slots in **shrouding strips,** and are riveted over to make a secure

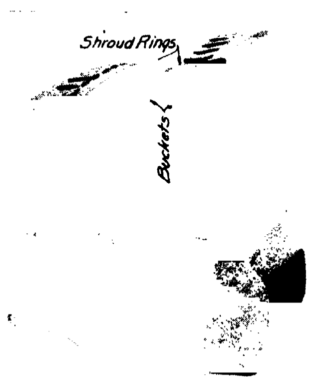


FIG. 367*a*. — Method of Assembling Buckets — Curtis Turbine.

FIG. 367*b*. — Curtis Turbine Wheel showing Bucket Assembly.

fastening, the method of attaching the buckets and shrouding being shown in Fig. 367*a* and *b* The spacing, or distance between the buckets, is fixed by a projection on the base of the bucket. The opening made for insertion of the buckets into the slot is filled by a spacing block after the last bucket is in place. Buckets are now made with slots instead of tenons. The stationary buckets are made like the moving buckets and are secured to segments which are bolted to the frame.

The plan and elevation, Fig. 368, of sections through two stages of this turbine show the general arrangement of nozzles and buckets. The first stage nozzles, Fig. 369, are formed in a plate attached to the casing by bolts, and extend only a short distance around the wheel rim. Because of increase in the volume of the steam as the pressure is lowered, the nozzle arc and the arc of the stationary reversing buckets, if used, are made progressively longer for the succeeding stages. In the lowest stage they extend entirely around the wheel. For this reason also, the height of the buckets is increased from stage to stage. The construction of the second-stage diaphragm and nozzles is shown in Fig. 370.

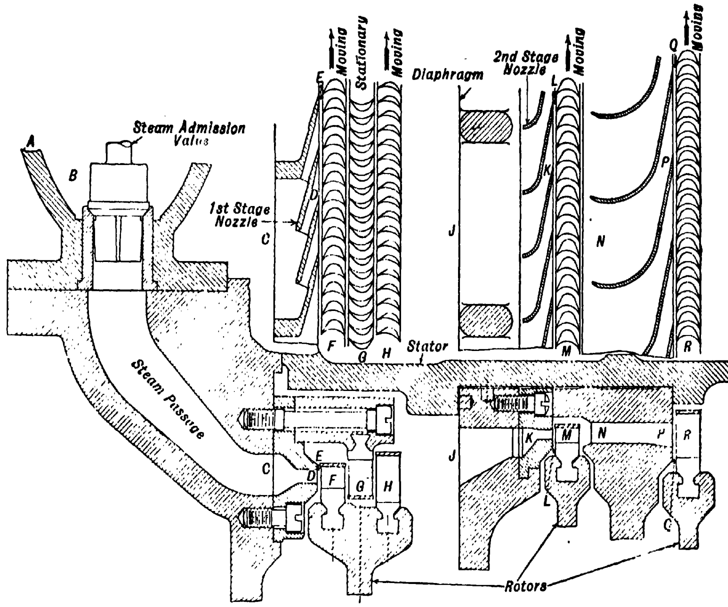


FIG. 368. — Diagrammatic Arrangement of Moving and Stationary Elements of Curtis Turbine.

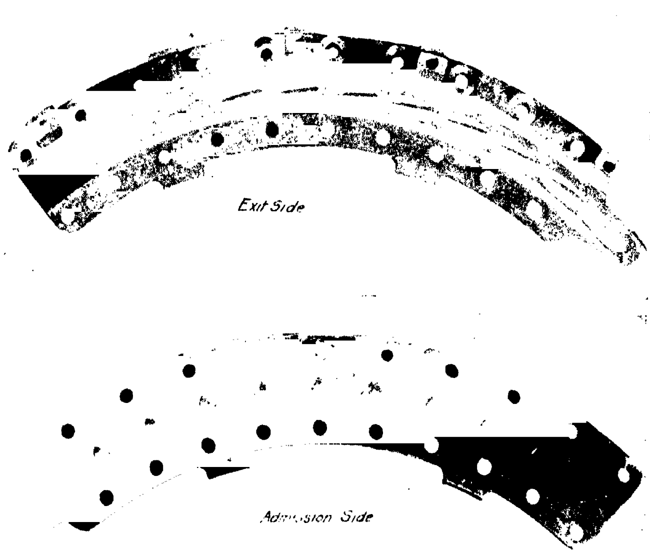


FIG. 369. — First Stage Nozzles Curtis Turbine.

An increase in height on the exit side of a bucket is necessary to allow for decrease in velocity of the steam. This also accounts for the increase in height of the buckets in the same and following stages.

The bearings have a shell divided horizontally into halves and lined with babbitt. A short length at the center of the shell is made spherical,

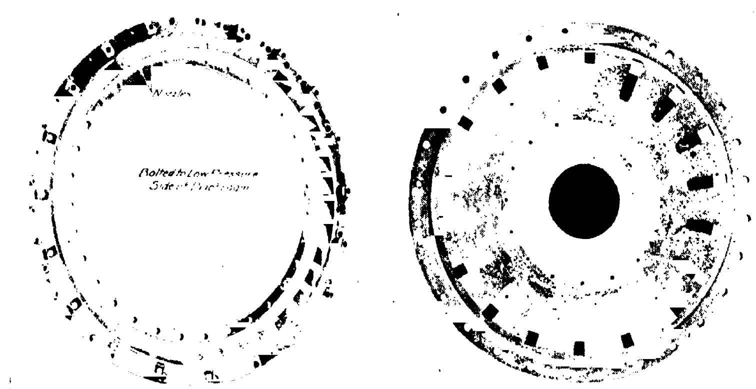


FIG. 370. — Second Stage Nozzles and Diaphragm, Curtis Turbine.

to fit into a corresponding seat in the pedestal, for purposes of alignment. The inner bearing, Fig. 371, shows in detail the arrangement of the parts. The upper shell is prevented from rotating by a screw bolt extending into

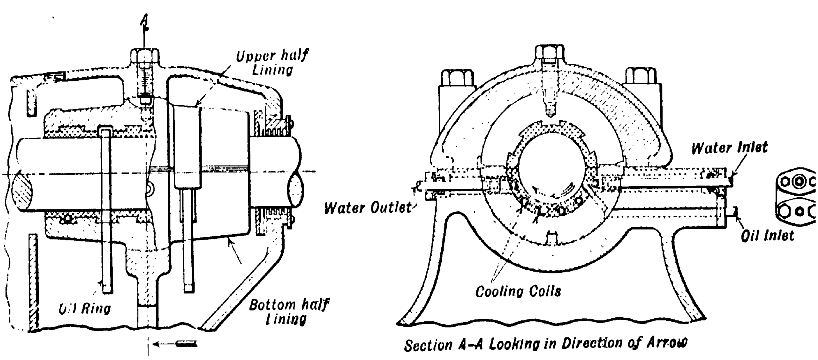


FIG. 371. — Section through Inner Bearing Curtis Turbine — Water Cooled.

a slot in the upper half of the shell. Oil under pressure is fed through piping to the lower half of the lining, and a cooling coil, through which water is circulated, lies in the lower half of the shell below the babbitt lining. The outer bearing carries a **roller thrust bearing** which controls

the position of the rotating parts. An enlarged view of a babbit thrust bearing is shown in Fig. 372 with the various parts named.

Leakage of steam out of the high-pressure casing or between pressure stages, or leakage of air into the low-pressure end, is prevented by either carbon or labyrinth packing. An assembly of the **carbon packing** is shown in Fig. 373. It consists of several segments with overlapping ends held

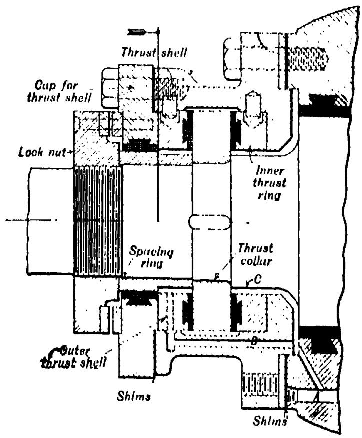


Fig. 372. — Details of Thrust Bearing used on Curtis Horizontal Turbines.

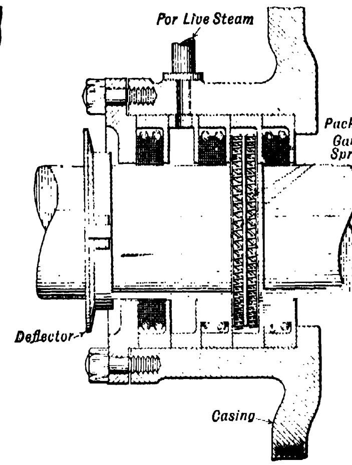


Fig. 373. — Carbon Packing.

between the partitions by a packing box, prevented from rotation and given a light bearing pressure on the shaft by **flat** or **garter springs.** To prevent entrance of air, steam at a pressure higher than that of the air is admitted as indicated.

The steam flows from the steam pipe through a strainer in the steam chest, and past the governor valve to the first-stage nozzles. After expansion in the nozzles, which results in high velocity, the steam strikes the first row of moving buckets on the rotor and gives up part of its velocity. It then passes to the stationary reversing buckets and so on through the velocity staging, to the second-stage diaphragm opening and into the second-stage nozzles, where it is again expanded and its available energy taken up by the buckets. This expansion and absorption are repeated for each pressure stage of the turbine.

457. Vertical Curtis Turbines. — Although this type of Curtis turbine is being superseded by the horizontal turbine, it is in satisfactory operation in many stations, in units as large as 15,000 kw. The general arrangement of a five-stage turbine is shown in Fig. 374, with the location of the governor, throttle valve, and admission valves marked. The **turbine shell**, or **casing,** which carries the stationary buckets, diaphragms,

and nozzles, rests on an exhaust base or a condenser, and supports above it the stationary part of the electric generator. The lower end of the vertical shaft rests on a step bearing, which carries the weight of the shaft, turbine wheels, and rotor of the generator. Two spherical bearings, one above the turbine and the other above the generator, keep the shaft in position. The **step bearing,** Fig. 375, consists essentially of two cast-iron blocks, the upper one **doweled** to the shaft, and the lower one supported by the base and adjusted by a large screw with a pipe passing through it to supply oil or water between the blocks. The lubricant, under a pressure of about 600 pounds per square inch, is forced out between the edges of the blocks and floats the shaft on a thin film. The turbine wheels consist of two steel disks riveted to a central spider carried by the shaft, each wheel carrying two rows of moving buckets.

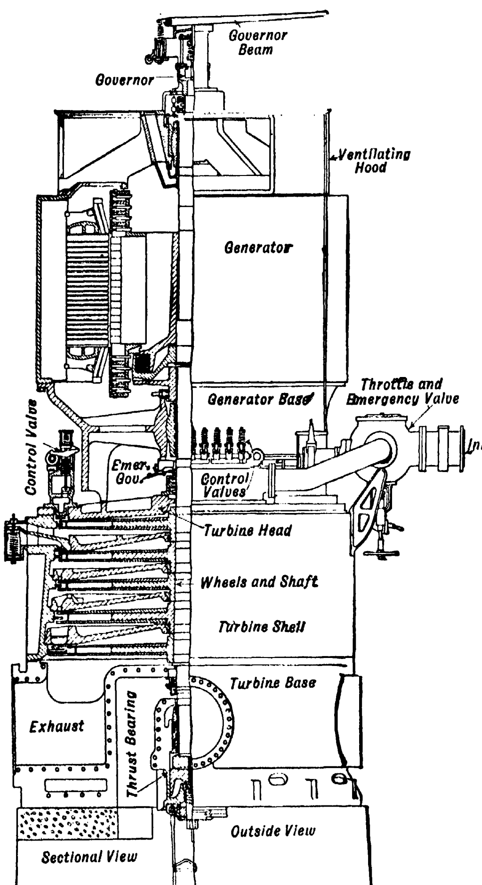


FIG. 374. — Half Cross-section of Vertical Curtis Turbine and Generator.

458. Governors and Valve Gears for Curtis Turbines. — Both centrifugal and inertia type governors are used on Curtis turbines, with three types of valve gears, namely, hydraulic or oil-operated, steam-operated, and mechanically operated. *In small units, a centrifugal governor mounted directly on the shaft controls the speed, by throttling the steam pressure. In large units, the speed is controlled by "cutting out nozzles" of the first stage.* By this method the amount of

steam is varied, while the pressure is maintained constant, thereby keeping a constant velocity in the nozzles and buckets.

459. Centrifugal Curtis Governor. — A sectional view, Fig. 376, of the centrifugal governor used with the hydraulic valve gear shows the weights, main spring, governor lever, and synchronizing springs, together with the other details. The weights, main spring, and connection rod are revolved by a vertical governor spindle, driven from the turbine shaft by a worm and worm gear. As the speed increases, the outward movement of the weights extends the spring, and the motion is transmitted by the **connection rod** to the governor lever, and thence through the valve gear to the admission valves.

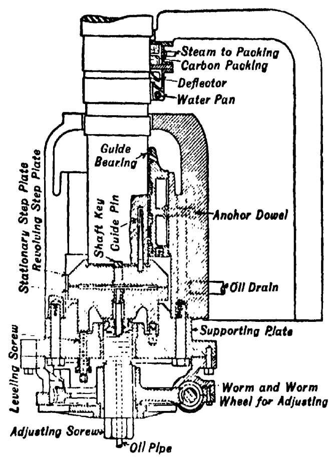


FIG. 375. — Step Bearing for Curtis Vertical Turbine.

The **auxiliary spring** is used to regulate the speeds through small limits, when synchronizing. Its tension is adjusted by means of a handwheel, worm and worm gear.

460. Inertia-type Curtis Governor. — This governor, Fig. 377, is used with steam-actuated and mechanical valve gears. Its power is derived from the centrifugal force and inertia effect of two **weight arms** pivoted on **ball bearings** and opposed by a

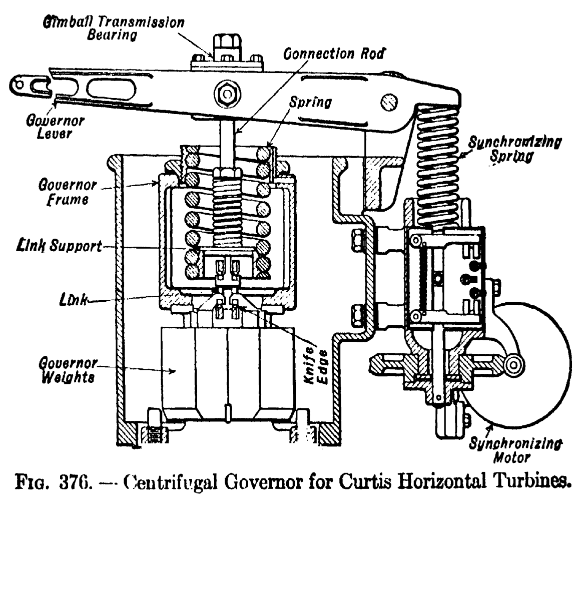


FIG. 376. — Centrifugal Governor for Curtis Horizontal Turbines.

helical spring. The movement of the weights, toward or away from the center of the driving spindle, produces a vertical motion, by a pair of **transmission links.** This vertical motion is transmitted through a spherical bearing to the governor lever and valve gear.

461. Curtis Hydraulic-operated Valve Mechanism. — This valve gear operates the steam valves by the movement of a piston in a cylinder, to which oil under pressure is admitted by a **pilot valve** controlled by the governor. The piston rod has a **rack** at its upper end which engages with a pinion on the end of a cam shaft. A series of cams and levers are arranged to raise the steam valves in succession as the cam shaft is turned.

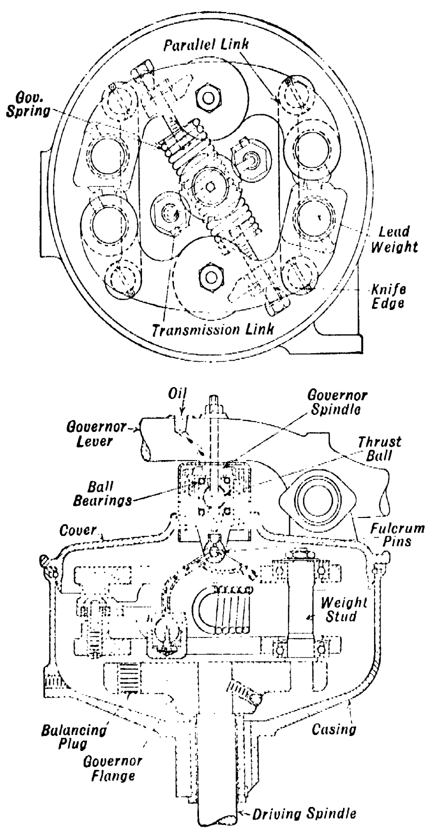


FIG. 377. — Assembly of Inertia Governor for Curtis Turbines.

The arrangement of the parts of this gear is shown in Fig. 378. As previously mentioned, a centrifugal governor is used. A gear oil pump, attached to the lower end of the governor spindle, furnishes oil under a pressure of 75 pounds per square inch, for the oil cylinder of the valve gear, and at 15 to 25 pounds per square inch, with the aid of a reducing valve, for the lubricating system. The **governor beam** is attached to one end of the **floating lever,** which is pivoted on a pin carried by the pilot valve stem, Fig. 379. The opposite end of the floating lever is connected by links to the piston rod of the oil cylinder. When the speed changes, the governor beam moves the pistons of the pilot valve away from their normal position, which is over the ports of the oil cylinder and oil is admitted under pressure to move the piston. This moves the rack and pinion and turns the **cam bar,** which operates the levers that raise or lower the steam valves. At the same time, the movement of the piston rod

returns the pilot valve to its central position, ready for the next change in speed.

If the speed of the turbine shaft drops, the governor beam raises the

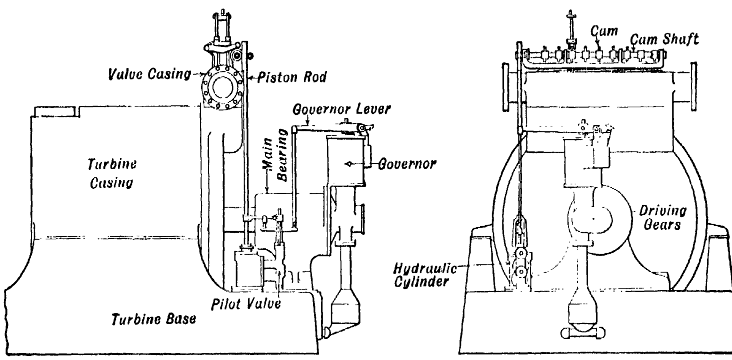


FIG. 378. — Arrangement of Hydraulic Valve Gear on Horizontal Curtis Turbines.

end of the floating lever, and the pilot valve stem admits oil above the piston. This lowers the piston, piston rod and rack, turns the cam bar and raises one or more steam valves. Lowering the oil piston also lowers the pilot valve to its stationary position.

462. Curtis Steam-operated Valve Gear. — The valve mechanism.

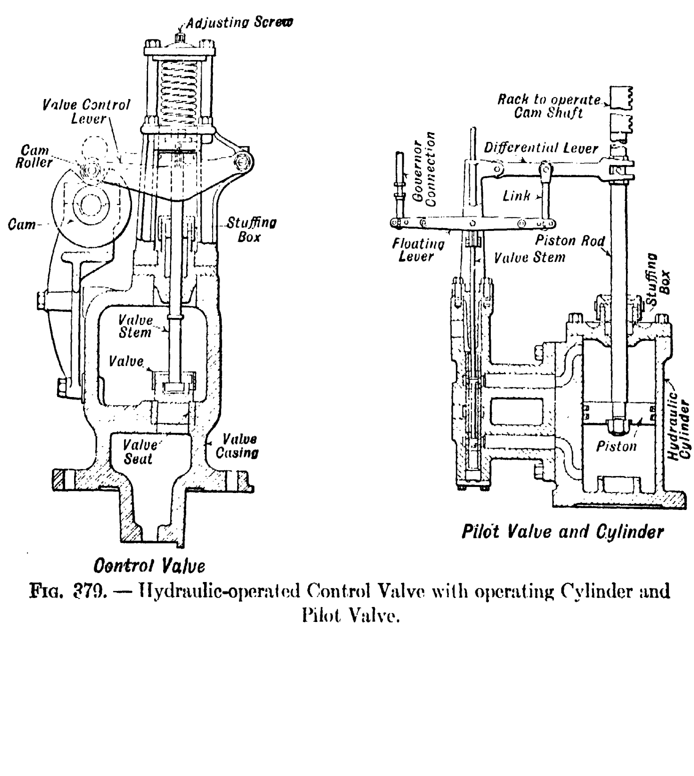


FIG. 379. — Hydraulic-operated Control Valve with operating Cylinder and Pilot Valve.

Fig. 380, consists of a steam cylinder with piston and pilot valve, a steam chest in which is a nest of double-beat poppet valves which are successively raised or lowered by hangers attached to a central valve stem connected to the piston in the steam cylinder, together with the lever and rods forming the connection to the governor, Fig. 381.

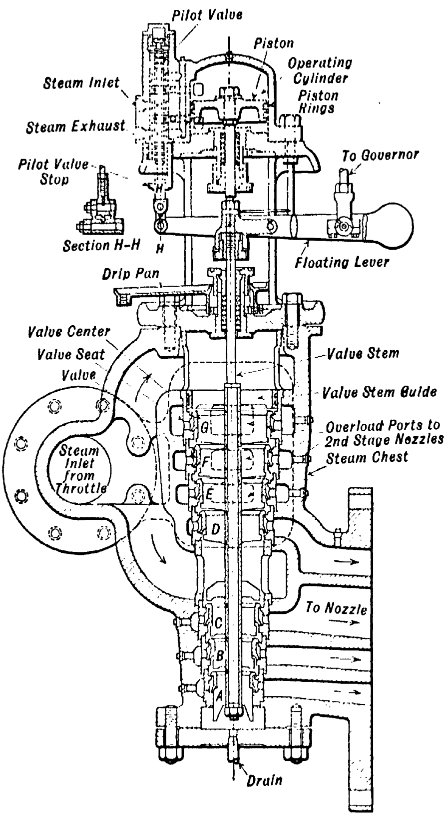


Fig. 380. — Steam-operated Valve Gear for Curtis Steam Turbine.

Steam from the throttle valve enters the steam chest through a cylindrical strainer and passes through several steam passages to the nozzles when the valves are raised. A separate hanger is used for each valve. The variation in clearances provided between valves and hangers allows the valves to be raised or seated in a definite order.

The steam poppet valves are all open when the turbine is started, and, on opening the throttle valve, steam is admitted to the lower side of the steam piston by the pilot valve. As the speed increases to normal, the governor transmission is raised, and the pilot valve is moved to such a position that steam is admitted above the operating piston. Movement of the operating piston seats the valves that are not required for the load, and, at the same time, the floating lever, with the connecting rod pin as a fulcrum, brings the pilot valve to its central position ready for another change in speed.

463. Curtis Emergency Governor. — On small Curtis turbines, two spiral clock springs, in tension, are carried by spring posts projecting from the face of a disk, Fig. 382, on the shaft. The free ends of the springs rest on pins near the edge of the disk. With an increase in speed of 10 per

cent above the normal, centrifugal force causes the end of the spring to leave the pin, and in so doing it strikes a trigger and releases a latch which holds the throttle valve open and allows the valve to close.

An emergency governor of the **double ring type,** Fig. 383, consists of two steel rings and a pin which passes through the shaft and normally

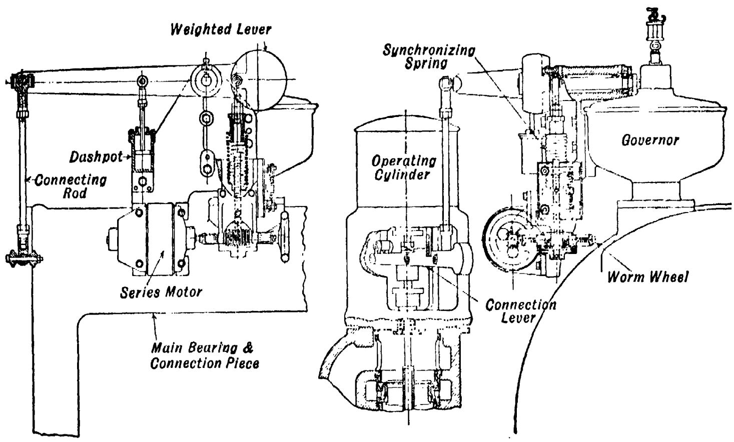


Fig. 381. — Assembly of Governor and Valve Gear Curtis Turbine.

supports them concentric with the shaft. A spring, in a box attached to the rings at one side, holds the pin against the shaft and maintains the

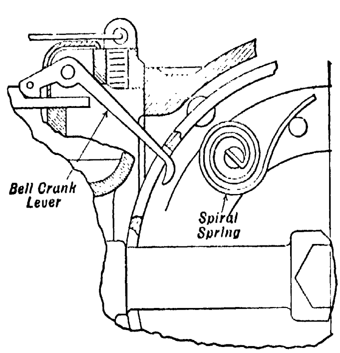


Fig. 382. — Curtis Spring Type Emergency Stop.

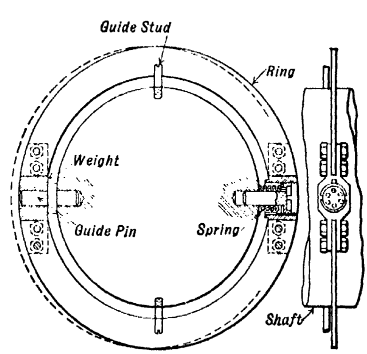


Fig. 383. — Curtis Ring Type of Emergency Stop.

normal concentric position. When the speed increases to an excess, centrifugal force overcomes the force of the spring, and the rings strike a trip, releasing the latch that holds the throttle valve open.

464. Throttle and Emergency Valve. — A Schutte and Koerting balanced throttle and trip valve, used on the larger Curtis turbines, is shown in Fig. 384, with the more important parts named. The **valve spool** and balancing piston form a single casting and provide a seat for the auxiliary valve attached to the valve stem. A stuffing box prevents leakage around the valve stem. Above the stuffing box, the valve stem is threaded through a nut which slides through the bracket on the valve yoke. A lever pinned to this sliding nut normally holds the valve open. With the valve closed, turning the handwheel, "to open", immediately opens the auxiliary valve, and steam above the balancing piston is discharged, thus relieving the pressure above the valve, which is now easily opened. The emergency governor releases the hook at speeds above normal, and the valve is automatically closed by the unbalanced pressure above the balancing piston.

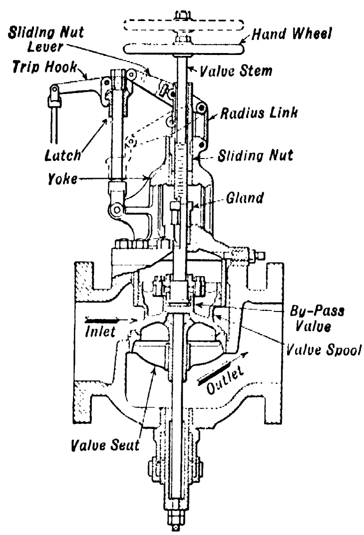


Fig. 384. — Assembly of Throttle Valve.

465. Kerr Turbines. — These turbines are of the impulse type, and, in the large sizes, use the Curtis and Rateau method of pressure-stage compounding. The " Economy " type has a casing built up of a number of circular sections parallel to the shaft. Between the steam-end section and the exhaust-end section are included the number of sections and the blading required by the steam conditions under which the turbine is to operate.

In the illustration, Fig. 385, there are four pressure stages. The first is a velocity section consisting of a nozzle ring, three rows of moving buckets, and two rows of stationary buckets, making use of the principle of velocity compounding as in the Curtis high-pressure element. The remaining pressure stages are of the Rateau type.

The sections are held together by bolts passing from the steam to the exhaust-end sections. The casing is split horizontally to give access to the blading, and is covered with a layer of non-conducting material surrounded by a sheet-metal covering.

The wheels are shrunk and keyed to the shaft, which is supported by two ring-oiled bearings. The buckets and method of governing are similar to the construction used in the small De Laval turbine. Carbon packing is used.

These turbines for land service are built in sizes up to 4500 brake horsepower for high-pressure condensing, non-condensing and bleeder service, and up to 2500 horsepower for low and mixed-pressure service.

466. Terry Turbine, Non-condensing. — This turbine is of the tangential-flow impulse type, using multi-velocity staging. The moving buckets are semicircular and milled in the solid rim of the wheel. The steam expanded in the nozzle enters the buckets at one side, and after leaving the buckets is re-directed on the wheel by a similar set of stationary reversing buckets. It is given several reversals before being discharged, as is shown diagrammatically in Fig. 386. By this method of using the steam, the speed of

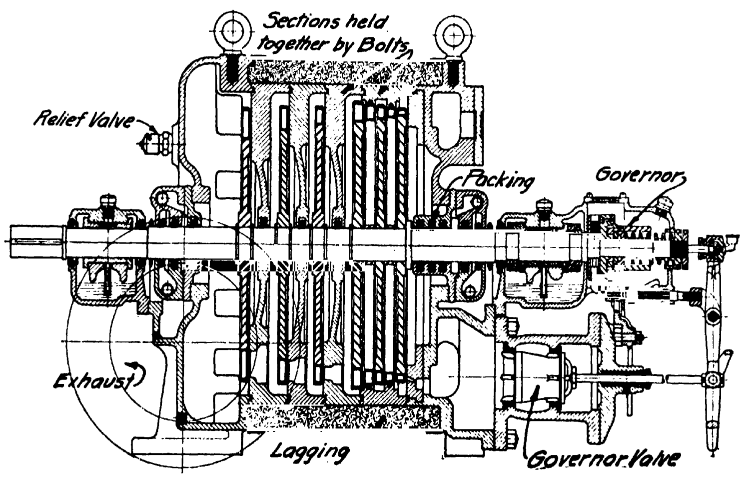


FIG. 385. — Kerr Economy Type Impulse Turbine.

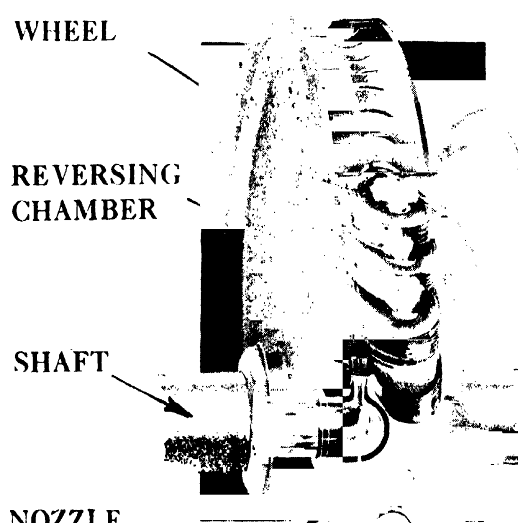


FIG. 386. — Nozzle and Wheel of Terry Turbine showing Action of Steam in Buckets.

rotation is reduced. The reversing buckets are arranged in several groups, at intervals around the wheel, and are bolted to the casing as illustrated in Fig. 387, where the top half is thrown back for purposes of inspection. The nozzles are separate from the reversing bucket and have hand-controlled valves to permit using only the number required by

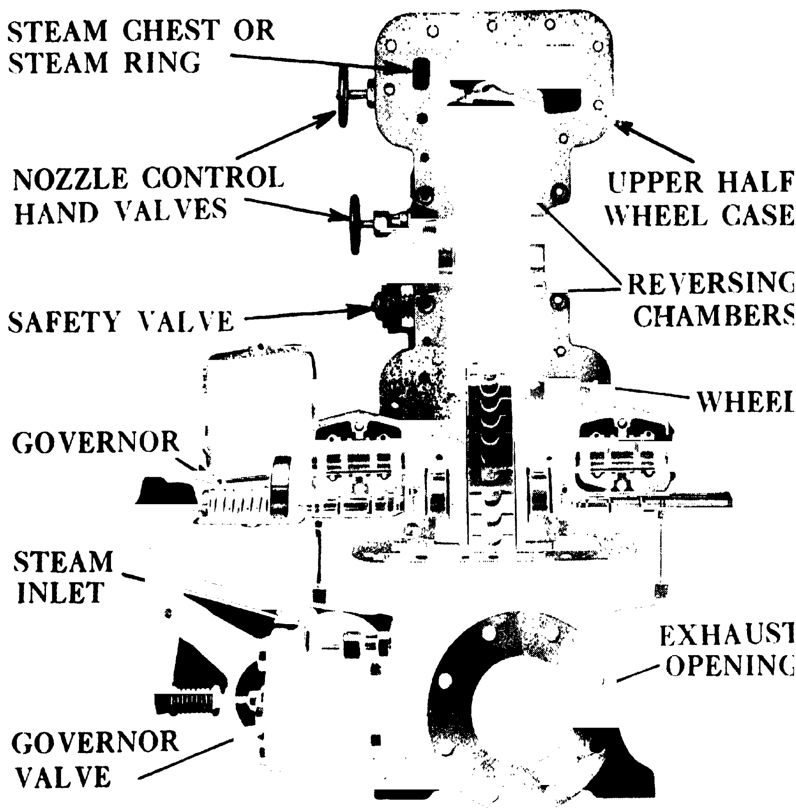


FIG. 387. — Non-condensing Terry Turbine.

the load and thus maintaining high efficiency. The speed is controlled by a centrifugal governor connected to a double-beat poppet valve.

The Terry condensing turbine uses the Curtis type of element in the high-pressure stages and the Rateau type of element in the lower-pressure stages. The governor used on the smaller sizes of this turbine is a centrifugal one, direct-connected to the governor valve, and on the larger sizes an oil relay system is used to operate the governor valve. Carbon ring packing is used to prevent leakage of steam in the smaller sizes, and labyrinth packing in the larger sizes.

A view of the **Sturtevant pelton multi-velocity type turbine,** Fig. 388, shows the arrangement of the nozzle and reversing buckets and the method of attaching them to the casing. The nozzle is made as a part of the re-

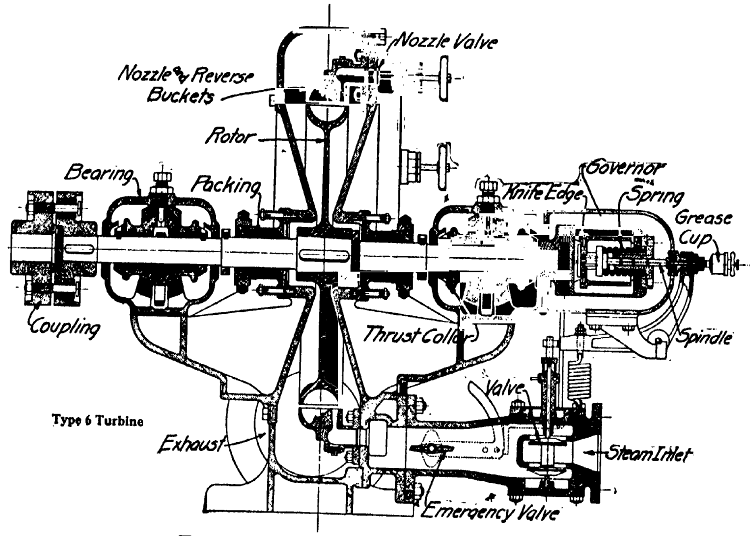


Fig. 388. — Sturtevant Impulse Turbine.

versing bucket casting, Fig. 389. The other parts of this turbine are similar to corresponding parts of the Terry turbine.

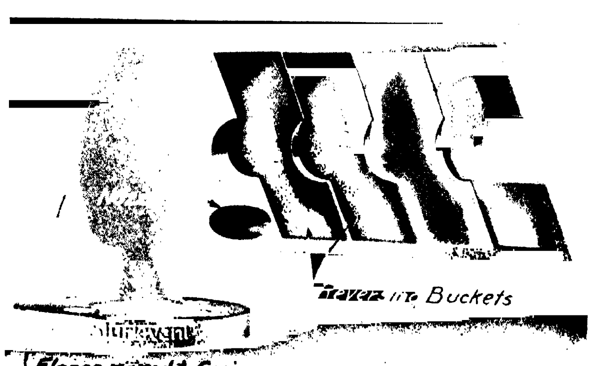


Fig. 389. — Sturtevant Turbine Nozzle and Reversing Buckets.

467. Westinghouse Impulse Turbine. — The construction, Fig. 390, used for the nozzles, blading, and reversing buckets enables units to be made as small as one kilowatt. The non-condensing unit has a single row

of moving buckets with a single set of reversing buckets arranged at the side of the wheel. By this method, the velocity of the steam is reduced in steps as in the velocity-stage Curtis turbine.

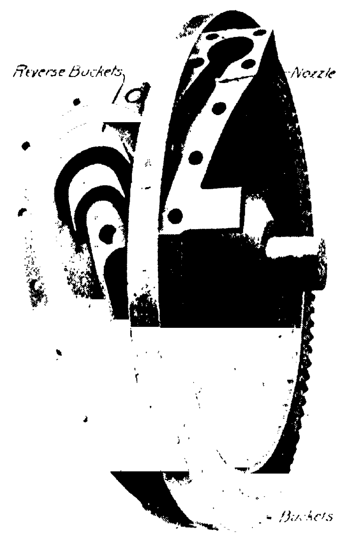


FIG. 390. — Westinghouse Impulse Turbine, — Casing Removed.

468. Elementary Theory, Impulse Turbine Nozzle. — The action of steam in a nozzle may be briefly described as follows: Steam supplied to the **bowl,** or entrance, of the nozzle, Fig. 391, passes through the **throat,** or smallest section, and expands through the conical divergent part of the nozzle, attaining a high velocity at the instant of discharge against the buckets. *The pressure of the steam at the throat drops to about* 0.58 *of the pressure in the bowl, and then decreases to the pressure of discharge at the* **mouth.** *The volume of the steam and its velocity increase from the throat to the mouth of the nozzle.*

The velocity at which steam is discharged from a nozzle can be calculated, since all the energy in a pound of steam in the bowl is present in the same pound passing out of the nozzle, neglecting the slight amount of heat lost by radiation during its rapid passage through the nozzle.

The total energy per pound of steam is the sum of its kinetic energy, $\frac{AV^2}{2g}$, and its internal energy, $H - APu$, in which V = velocity in feet per second, A = reciprocal of the heat equivalent of work = 1/778, H = total heat per pound of steam, and APu = the external work performed, as explained in Art. 82, page 91. Denoting the bowl and mouth condition respectively by subscripts 1 and 2, there results, for saturated steam,

$$\frac{AV_1^2}{2g} + H_1 - AP_1u_1x_1 = \frac{AV_2^2}{2g} + H_2 - AP_2u_2x_2$$

Since V_1 is very small compared to V_2, and the difference in the external work items is small, they may be neglected, and the equation becomes

$$\frac{AV^2}{2g} = H_1 - H_2,$$

or

$$V_2 = \sqrt{2 \times 778 \times 32.2\,(H_1 - H_2)} = 223.8\sqrt{H_1 - H_2} \qquad (99)$$

in which, H_1 and H_2 equal the total heat per pound of steam at the bowl and mouth pressures respectively, taking into account the quality of the steam at those points. Since the passage of steam is almost instantaneous, the change in pressure and volume is assumed to be adiabatic, neglecting radiation. The quality after expansion can be calculated with this assumption, or the value H_2 can be found by using the heat-entropy chart.

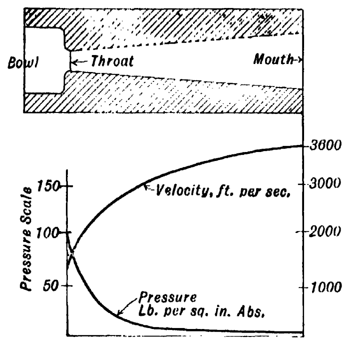


FIG. 391. — Curves showing Pressure and Velocity Changes in an Expanding Nozzle.

The actual conditions, however, do not give an adiabatic change on account of the friction of the steam against the surface of the nozzle. The energy lost in friction is given back to the jet of steam in the form of heat, so that the steam at the mouth of the nozzle has more heat per pound than it would have if the expansion were adiabatic. If the effect of friction in decreasing the velocity of the jet is y part of the theoretical heat energy drop, then the velocity, considering friction, becomes

$$V_2 = 223.8 \sqrt{(H_1 - H_2)(1 - y)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

The value of y varies with different conditions. With the nozzle shown it is about 10 per cent for steam discharged to atmospheric pressure.

Example 52. — The steam pressure in the bowl of a nozzle is 100 lb. per sq. in. abs., and the vacuum at the exit is 28 in. of mercury, barometer 29.92. Neglecting friction, find the theoretical velocity of the jet of steam leaving the nozzle, if the initial condition of steam is dry.

Solution. — Using Equation (100) and Steam Table, page 89.

$$\text{Velocity in feet per second} = 223.8 \sqrt{H_1 - H_2}$$
$$= 223.8 \sqrt{1188.4 - 900.2} = 3088$$

H at 100 lb. per sq. in. abs. $= 1188.4$

$H_2 = x_2L_2 + h_2$ at 1.92 in. mercury absolute $= 0.803 \times 1036.6 + 67.8 = 900.2$

From Art. 100 $\theta_1 + \frac{x_1L_1}{T_1} = \theta_2 + \frac{x_2L_2}{T_2}$, or $x_2 = \left[\theta_1 - \theta_2 + \frac{xL_1}{T_1}\right] \div \frac{L_2}{T_2}$

Referring to *steam tables*, page 89, for entropy values

$$x_2 \text{ at 1.92 in. mercury abs.} = \frac{0.4736 - 0.1292 + 1.1309}{1.8489} = 0.803$$

469. Elementary Theory, Impulse Turbine Buckets. — The kinetic energy of the steam jets leaving the nozzles is changed into energy of rotation in the buckets by the direct action or impact of the steam, and by the reaction of the steam leaving the buckets after being deflected in its course.

Consider a turbine wheel to be running with a velocity at the center of the buckets of u feet per second. Steam enters the buckets with velocity V, Fig. 392, at the absolute entrance or nozzle angle α. Relative to the bucket, steam enters at angle β and with velocity W. It is deflected by the bucket and leaves at an angle β_1 and with a velocity W_1 relative to the bucket. While in the buckets it has the velocity of the buckets, u. Its direction and velocity of exit with regard to a stationary point, that is, its absolute exit angle and velocity, are found by combining the velocities W_1 and u, giving V_1 as the exit velocity. Work has been done by the change in velocity of the steam. On striking the buckets the steam gives to the buckets its component of velocity in direction of rotation, bd. On leaving the bucket the steam emerges with an absolute velocity V_1 found as previously explained, of which gf is the component in direction opposite to the direction of the bucket. Since the reaction due to exit is equal to the action but in opposite direction, gf is the useful change in velocity. The sum of bd and gf is the complete useful change in velocity of the steam, or acceleration, imparted to the bucket wheel. Since the force, F, equals the mass, M, times the acceleration, a:

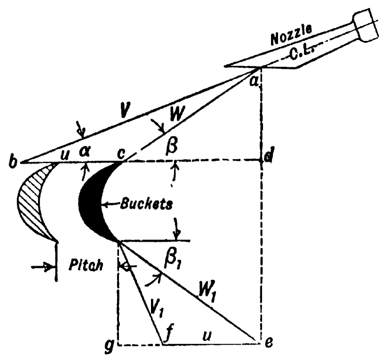


FIG. 392. — Velocity Diagram for Impulse Turbine Buckets.

$$F = Ma = \frac{1}{g}(bd + gf) \text{ per pound of steam} \quad . \quad . \quad . \quad . \quad (101)$$

and since the force was exerted through a distance, S, equal to the velocity (u) of the bucket, or (bc), in feet per second

$$F \times s = \frac{(bd + gf)}{g} bc \text{ foot pounds per second} \quad . \quad . \quad . \quad . \quad (102)$$

To find the work done, V and u are laid out to scale at the angle α and, after the construction is completed, the values of bd and gf are measured to the same scale.

The significance of the terms **absolute** and **relative,** as applied to the steam entering and leaving the buckets of a turbine, may be made clearer by considering the bucket to be a boat moving in a stream at u feet per second. A person on the bank would throw a ball to a person on the boat by aiming ahead of the boat. The actual or absolute direction of motion of the ball would make the angle α with the center of the stream. The per-

son on the boat would receive the ball at the angle β. If the ball were allowed to rebound to the opposite side it would leave at the angle β_1 and with a velocity W_1, relative to the boat; but since the person on the boat is moving at a velocity u the ball would have an actual or absolute angle of leaving greater than β, and an actual velocity V_1 at the opposite shore.

470. Reaction Turbines. — The best-known reaction type of turbine is the Parsons steam turbine, introduced in the United States in 1895 and

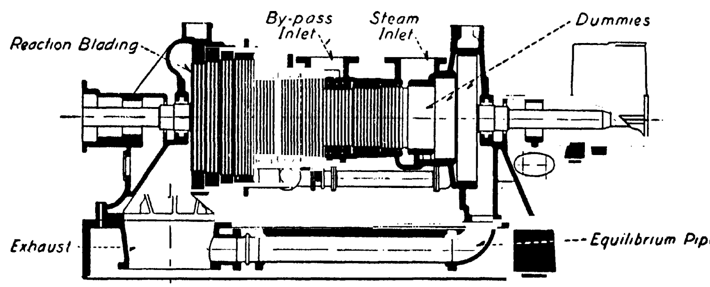


FIG. 393. — Westinghouse Single Flow Reaction Turbine.

manufactured, by the Westinghouse Electric and Mfg. Co., as the Westinghouse-Parsons turbine. The general arrangement of this turbine and its later forms are shown in the diagrams in Figs. 393 to 395.

1. *Single-flow turbine.* — The general construction may be likened to an expanding nozzle made up of a number of transverse sections, alternate sections being carried by a shaft through the center, and the other stationary sections supported by the outside or frame of the expanding nozzle. Each stationary section consists of a row of radial buckets inserted into a dovetailed, circumferential groove in the frame or cylinder, or into blade rings carried by the cylinder. The rotating rows of buckets are carried by comparatively thin cylindrical drums having similar dovetailed grooves in their circumferences, in which the blading is securely fastened, Fig. 393. Steam, after passing the emergency and governor valve, enters an annular chamber in the turbine casing and passes through the stationary and moving buckets, which are made of increasing length to allow the necessary area for the steam to pass, until it reaches the exhaust connection. **Dummy pistons,** connected with **equilibrium pipes** to the exhaust spaces of the second and third drums, are provided to equalize the unbalanced end-thrust along the shaft, caused by the passage of the steam. The dummy pistons are usually grooved to join with corresponding projections on the casing. This forms a labyrinth packing and minimizes the leakage of steam around the balance piston.

2. *Single-flow, impulse and reaction turbine.* — An impulse element in

this turbine, Fig. 394, replaces the high-pressure drum in the single-flow reaction turbine, and generally shortens the machine. A dummy piston is used to balance the axial thrust.

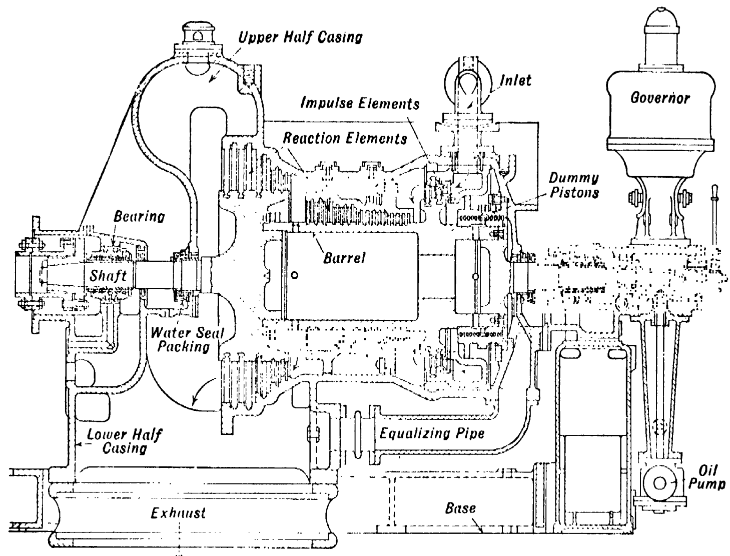


FIG. 394. — Single Flow-Impulse and Reaction Turbine.

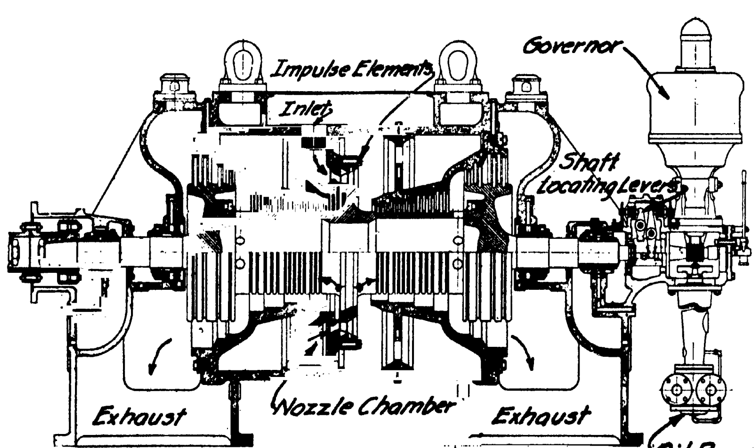


FIG. 395. — Westinghouse Double Flow Reaction Turbine.

3. *Double-flow, impulse and reaction turbine.* — The nozzle chamber in this turbine, Fig. 395, is arranged to deliver high-pressure steam to a two-velocity-stage impulse wheel, from which the steam divides and passes

toward the two ends of the machine through the reaction-type blading. This is, in effect, two single-flow reaction turbines, no dummy pistons being required to balance the axial thrust. A unit of about double the power is obtained at the same speed. To obtain the increased capacity in a single-flow machine, under the same conditions, the length of the buckets would have to be increased, thus increasing the mean diameter of the blade ring

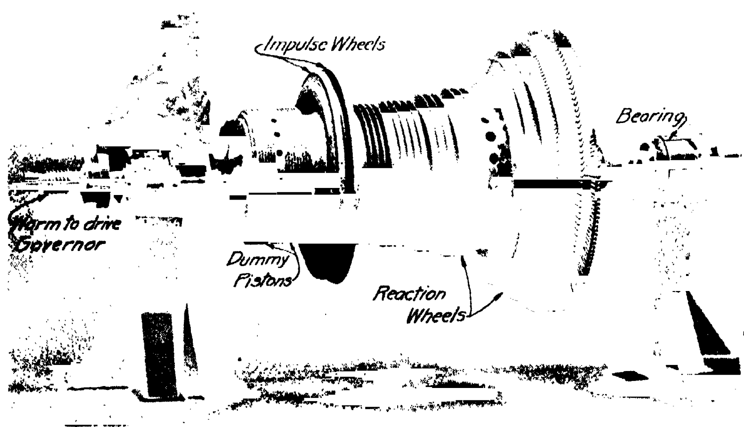


FIG. 396. — Rotor for Combined Impulse and Reaction Single-flow Turbine.

and the stresses caused by centrifugal force. The machine may be regarded as analogous to the combination of two multiple-expansion reciprocating steam engines working on the same shaft.

4. *Semi-double flow, impulse and reaction turbine.* — In this type of turbine, steam enters the impulse element, passes to a small drum, and divides, going in opposite directions through the buckets of the large drums. A dummy ring is used to balance the end-thrust produced by the intermediate single-flow section. This design may be considered analogous to a four-cylinder triple-expansion reciprocating engine, having high pressure, an intermediate pressure and two low-pressure cylinders.

The above rotors consist of hollow steel drums, finished inside and outside and fastened to flanged or enlarged ends on the shaft. The drums of larger diameter may consist of separate steel rings balanced and pressed on the central drums. The end-thrust of a single-flow turbine was balanced by steam pressure against dummy pistons in the early type of turbine. The use of the Kingsbury thrust bearing has made it possible to do without the two larger dummy pistons shown in Fig. 393. In the double-flow rotors, end-thrust is balanced by the action of the steam; in turbines combining

impulse, or single-flow sections, with double-flow sections, dummy pistons or rings are used. A rotor for the combination impulse and reaction single-flow turbine is shown in Fig. 396.

The method of securing the impulse buckets in the cylinder and rotor is shown in Fig. 397. The purpose of the compound wedges is to allow the

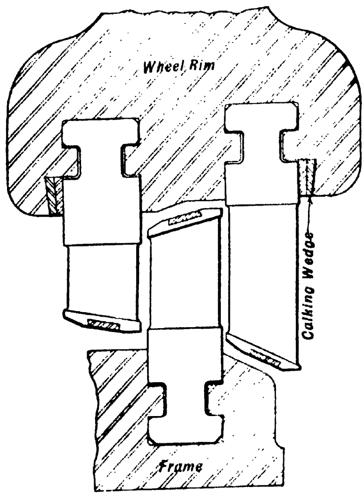


FIG. 397. — Method of Installing Impulse Blading.

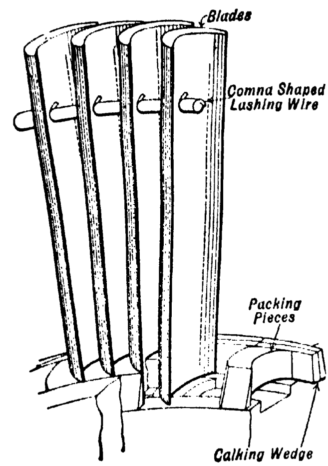


FIG. 398. — Method of Assembling Reaction Blading.

buckets to be properly and securely fixed in position on the rotor without the necessity of extreme accuracy in machine work required for a piece filling the slot. The reaction buckets, Fig. 398, are made of phosphor bronze, heat treated, and formed with a shoulder on one side at the root end. At the bottom of the dovetail grooves, on the rotor and cylinder, a rectangular slot is milled to receive the shoulder on the buckets. Steel distance pieces with beveled edges are fitted into the dovetailed grooves above this projection on the buckets, thus holding them securely. The buckets are made thicker at the root, and tapered to the standard section to increase strength and distribute vibration. With the larger sizes of buckets, steel wedges are calked into the space between the buckets and packing pieces and the side of the dovetailed groove.

To support the outer ends of the buckets and to maintain a proper distance between them, a **lashing wire** of " comma " section is passed through holes of similar section punched in the buckets. After the buckets are straightened and spaced, the tail of the comma is bent over. The shape of the reaction bucket is different from that of the impulse bucket, because it is shaped to allow the steam to expand.

In the larger slow-speed turbines, the bearings are lined with babbitt

and made in halves. They are provided with pads resting in the spherical bore of the pedestal, to make them self-aligning. For high speeds, the bearings are made with special construction to allow some flexibility, on account of the tendency of the spindle and drums to rotate about the gravity axis instead of the center line of the spindle at speeds above the

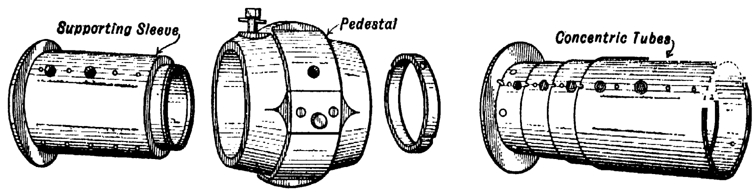


Fig. 399. — Tubular Type of Turbine Bearing.

critical speed. The bearing then consists of several concentric tubes having a small clearance between them, the clearance space being filled with a film of lubricating oil. This set of tubes, Fig. 399, is carried by a cast-iron supporting sleeve provided with four steel pads resting on the spherical bore of the pedestal. The surfaces of these blocks conform to their support and provide for self-alignment, as in the case of the plain bearing.

At the ends where the shaft passes through the casing, **water-seal packing,** Fig. 400, prevents the escape of steam from the high-pressure end or entrance of steam into the low-pressure end. Packing by this method is accomplished by means of a disk or impeller, revolving with the shaft in a small recess in the turbine casing to which water is supplied. Centrifugal force carries the water to the outer edge of the disk, and the water effectually prevents the passage of air or steam.

For marine service, at speeds below that required to produce a centrifugal force sufficient to maintain the water seal, the governor is arranged to operate a relay, turning water off and steam into the packing chamber; and a labyrinth packing, located inside the water-seal packing, is added.

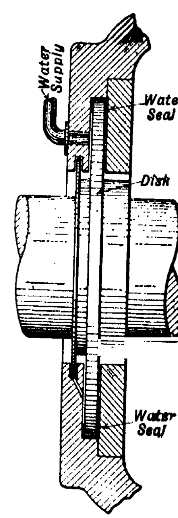


Fig. 400. — Water-seal Packing.

In the modern turbine the end-thrust caused by the difference in pressure at the entrance and exit of the several stages is taken by a thrust bearing, preferably of the **Kingsbury type.** In this bearing, Fig. 401, several **pivoted segments,** or **shoes,** receive the pressure of the **thrust collar** through oil films of wedge section, the thicker part of the film being

where the oil drawn in by rotation enters the space between segment and collar. Pressures as large as 350 pounds per square inch for moderate speeds, and 500 pounds per square inch for high speeds, are successfully carried, as compared with 50 pounds per square inch which is a high value for the ordinary thrust bearing with parallel contact surfaces.

The axial position of the rotor is fixed accurately by means of an **adjusting bearing** shown at the right-hand end of the shaft, Fig. 395. It consists of a number of collars on the shaft, into which fit brass rings fixed in two adjustment blocks. The upper and lower blocks, which bear on opposite sides of the collars, are moved by **micrometer screws,** thus permitting the axial position to be accurately known.

471. Multiple-cylinder Turbines. — In this type of turbine, the steam is expanded successively in more than one turbine shell. *The cross-compound turbine has the high-pressure and low-pressure elements side by side.*

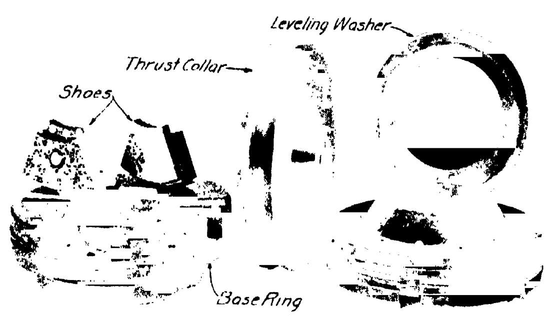


FIG. 401. — Kingsbury Thrust Bearing.

In the 74th St. station of the Interborough Rapid Transit Co., New York, the 30,000 kilowatt turbine unit has a high-pressure element running at 1500 r.p.m., and a low-pressure element at 750 r.p.m., each direct-connected to a generator at 25 cycles.* *In a tandem-compound turbine, the expansion takes place in two turbine shells, in tandem.* Such a turbine unit, in the Northwest station of the Edison Company of Chicago, has a capacity of 35,000 kilovolt amperes at 1200 r.p.m. and 60 cycles.

472. Governors and Valve Gears for Reaction Turbines. — The governors used on Westinghouse turbines are of the centrifugal type. They control the admission of steam by three types of valve gear, namely, (1) by direct lever connection, (2) by steam-operated relay, (3) by hydraulic or oil-operated relay.

* An alternating current is said to complete a cycle when it rises to a maximum in one direction, falls to zero, and then rises to a maximum in the opposite direction. A 25-cycle current would reverse its direction of flow twenty-five times a second.

473. Governor with Direct Lever Connection.—This gear is used on the smaller reaction turbines. An illustration of one governor of this type, Fig. 402, shows the weights, levers, springs and the lever connections to the admission valves. The governor spindle is vertical and is driven from the

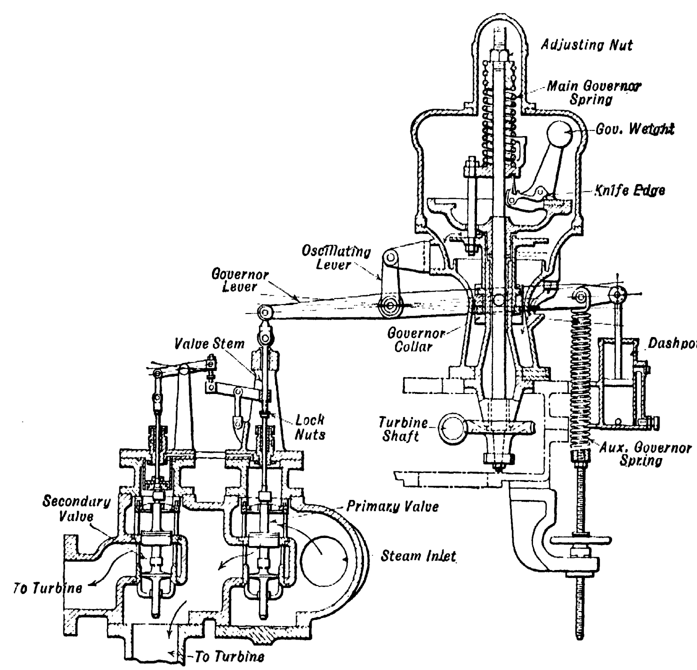


FIG. 402. — Governor and Direct Connected Valve Gear for Westinghouse Turbine.

turbine shaft by a worm and wheel. The principal adjustment for speed is made by turning the nut at the top of the spindle to control the initial compression of the spring which surrounds the spindle. An auxiliary spring is provided to permit more sensitive adjustment.

In the position shown, the speed corresponding is a maximum, the governor collar to which the main lever is connected is in its highest position, the connecting levers are in their lowest position, and the valves are closed. As the speed decreases the force of the spring overcomes the centrifugal force produced by the weights, and they approach the spindle. The collar is lowered, thus raising the opposite end of the main lever, which has a fulcrum that swings about a point on the frame and raises the admission valves.

The two **double-beat poppet valves,** called " primary " and " secondary " valves, are raised successively by the lever connections. The primary

valve admits steam to the first stage of the turbine blading for normal load conditions. The secondary, or overload, valve remains seated until the primary valve has been raised to give full port opening. Then, for overloads, the hanger for the secondary valve strikes the lock nuts on the valve stem and raises the valve to admit high-pressure steam directly to a lower stage.

474. Steam-operated Relay. — This gear, which admits steam in a succession of puffs, was used on the early types of Westinghouse-Parsons turbines, up to about 1909, and is still used abroad. As the sizes of turbines increased, the effect of reaction and vibration, caused by the pulsating movement of the steam, caused it to be discarded in favor of the oil relay mechanism, which maintains a practically constant flow of steam. However, the steam relay is still giving satisfaction in the older machines and a brief description of it will be given. The same sizes of turbines are now equipped with direct-connected governors.

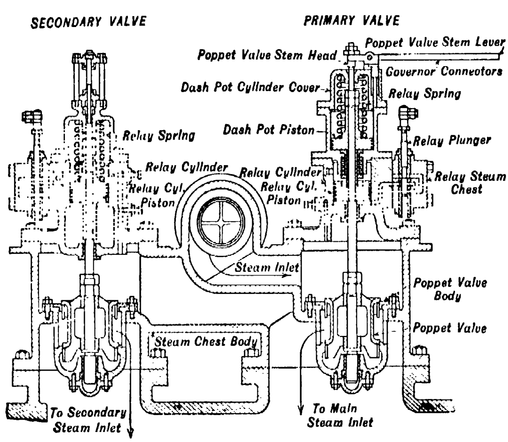


FIG. 403. — Steam-operated Relay Valve Westinghouse Turbine.

The double-beat poppet valve, Fig. 403, that admits steam to the first drum is operated by the steam piston in a cylinder above the steam chest. Steam is constantly supplied to the space below the piston through a steam passage from the steam chest, and by its pressure raises the piston and valve against the resistance of a heavy spring above the piston. A pilot valve, under control of the centrifugal governor, opens and closes an exhaust port to the cylinder and thus the pressure below the piston alternately decreases and increases. This lowers or raises the admission valve and admits or cuts off steam from the port. The reciprocating movement of the pilot valve is derived, through the system of levers, from an eccentric operated by a worm and worm wheel from the shaft. The fulcrum of the main lever is carried by the collar of the governor, and the amount of the reciprocating movement given to the pilot valve is controlled by the position of the governor collar.

With increasing loads, the slight reduction in speed lowers the fulcrum

of the lever, the pilot valve keeps the port closed longer, and the steam admission valve is open longer, until with full load the admission of steam is almost continuous. The steam is thus admitted in puffs at full pressure, the length of the puff being controlled by the governor.

At overloads, another pilot valve, controlled by the governor, causes an auxiliary admission valve located at the second drum to admit high-pressure steam to that stage.

475. Hydraulic or Oil-operated Relay. — For the larger turbines, the oil relay mechanism is used to control the steam admission valves. Two double-beat poppet inlet valves are used, Fig. 404, as with the steam relay mechanism, one admitting steam from the throttle valve to the first row of blading or to the high-pressure nozzles, depending on the construction of the turbines, and the other admitting additional steam, if required, to another set of nozzles or to a lower stage of blading.

The governor spindle is revolved by connection to the turbine shaft, through a worm and worm wheel. The **floating lever** connected with the governor collar has a fulcrum swinging by a link attached to the frame. This lever is constantly oscillated by a cam and roll, in order to overcome friction and allow the linkage to come to a new position more readily. By the slight movement given to the pilot valve, alternately admitting oil above and below the relay piston, that piston is kept oscillating about $\frac{1}{10}$ of an inch.

When the collar of the governor is lowered by decrease in speed, the pilot valve is raised, admitting oil at 50 pounds pressure above the relay piston, which thus moves downward and raises the primary steam admission valve. The pilot valve is returned to its mid-position by the movement of the relay piston.

If the relay piston is lowered still further, the lever above the secondary or overload valve, which thus far moved freely in the hanger on its valve stem, comes against an adjustable stop and raises the valve.

In case of emergency or overspeed, if the emergency governor is tripped, steam is admitted into the controlling steam cylinder and forces down the steam piston. The auxiliary pilot valve is forced down, admitting oil below the relay piston, which rises, closing the admission valves regardless of the governor position.

476. Action of Steam in the Buckets. — In the reaction turbine, steam expands continuously in the stationary vanes and moving buckets. Its velocity and volume increase, requiring increased area between the buckets. As it is inconvenient to make a large number of different diameter drums and lengths of buckets, a small number of drums are used, with buckets of one length on a drum.

Steam is directed against the moving buckets by the stationary guide vanes, with velocity V, at an angle α, as represented in Fig. 405. If the

bucket velocity is u feet per second, the relative entrance velocity is W. The steam expands between the moving buckets, increasing its velocity

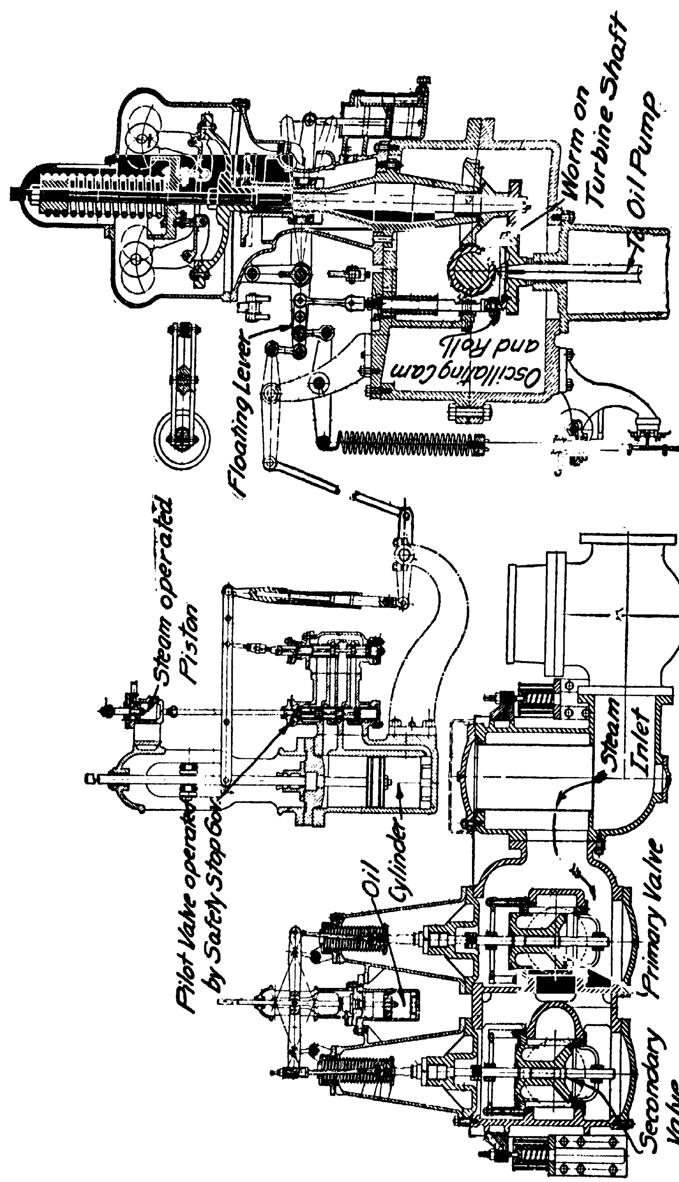


FIG. 404. — Oil-operated Relay Valves controlled by Centrifugal Governor.

and leaving with relative velocity W_1 and exit angle β_1, approximately equal to its entrance velocity V and entrance angle α. Compounding

the relative velocity W_1 with the velocity u, which it has while on the buckets, the absolute exit velocity is V_1. The steam entering the next set of stationary vanes with this velocity is further expanded and is redirected against the next row of moving buckets with a slightly increased velocity.

The work done per pound of steam per stage is found by laying out the velocities and angles as in Fig. 405, and measuring the components of the

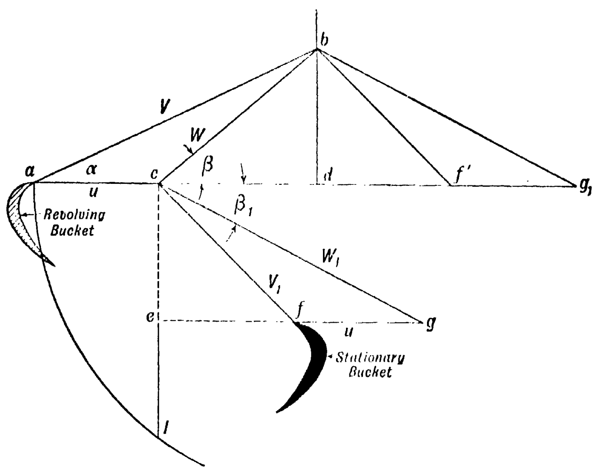


FIG. 405. — Velocity Diagram for Reaction Buckets.

absolute velocities V and V_1 in direction of rotation, that is, $(ad + ef)$; and the work per pound of steam, F_s, is as follows:

$$F_s = \frac{1}{g}(ad + ef)\,ac \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (103)$$

A simple geometrical construction in this case shows that $(cl)^2$ is equal to $(ad + ef)\ ac$, since $ad + ef = cg$ by construction.

477. Low-pressure, Mixed-pressure, and Bleeder Turbines. — The general construction of low-pressure, mixed-pressure, and bleeder turbines is the same as that of the standard forms, except for the special valves and connections required.

Low-pressure turbines use exhaust steam from engines, pumps, or other steam-driven apparatus. The exhaust steam is generally supplied at slightly more than atmospheric pressure and expanded to as low a pressure as the condensing apparatus and conditions will permit.

If the steam is taken from non-condensing engines, the additional power developed may be 80 to 100 per cent of that developed by the engines. If the engines have been running condensing, the use of low-pressure turbines may increase the power from 25 to 40 per cent.

These results are obtained because the steam is expanded to a lower pressure, before leaving the turbine buckets, than is practical in the reciprocating engine, on account of the large size of the low-pressure cylinder that would be required. Also, the condensation of entering steam caused by contact with cooler surfaces, as in the engine cylinder, is decreased, since the flow of steam is continuous in the turbine.

The **mixed-pressure turbine** is a low-pressure turbine provided with special nozzles automatically supplied with high pressure steam which is

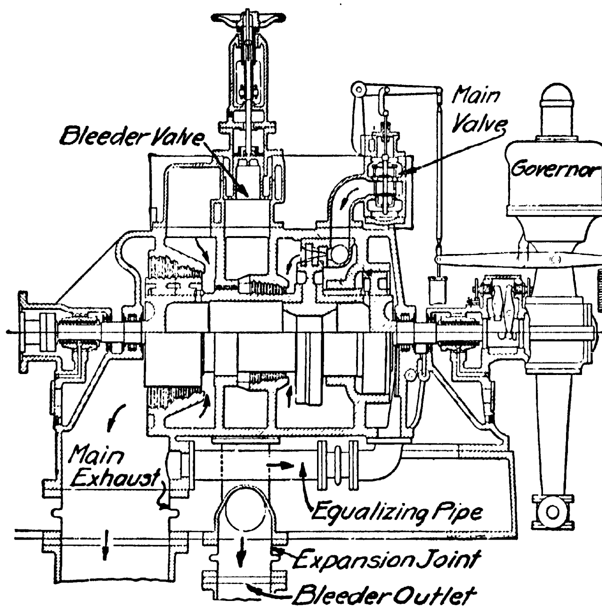


FIG. 406. — Section through a Westinghouse Bleeder Type Turbine.

completely expanded in the turbine in case the low-pressure steam is insufficient in amount to maintain the speed. It may carry all or any part of its rated load on high-pressure steam.

Bleeder, or extraction, turbines are used in some industrial plants where there is a demand for medium- or low-pressure steam for process work or for heating. Provision is made for this requirement by bleeding steam from an intermediate stage of the turbine, Fig. 406, through a weighted valve which controls the pressure and amount of steam taken away. The remaining steam passes on through the turbine to the exhaust.

478. Lubrication of Steam Turbines. — The bearings of large turbines and gears are supplied with oil by gravity or by forced feed. Ring-oiled bearings are also used on both large and small turbines.

The **gravity-feed system** for a De Laval marine turbine and gears, Fig. 407, consists of an elevated tank from which oil is supplied to bearings and gears at about 5 pounds per square inch pressure. The oil, after leaving the bearings, collects in a drain tank from which it is drawn by a separate

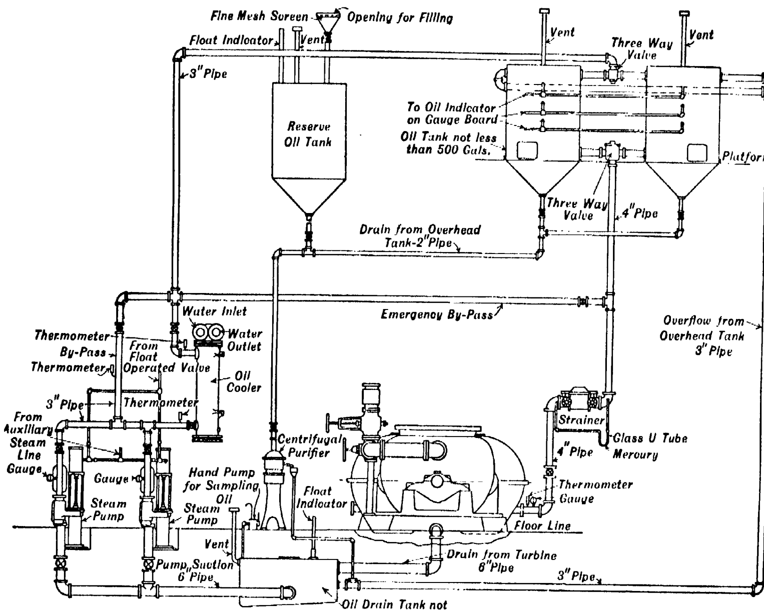


FIG. 407. — Oiling System for a Steam Turbine, including Oil Pump, Oil Cooler, Strainer and Centrifugal Purifier.

pump. It is pumped through a coil in a water-cooled tank and thence to an elevated tank from which it flows through a filter to the bearings. A **centrifugal purifier** is used to keep the oil free from impurities.

In the **forced-feed system,** Fig. 408, a rotary oil pump, driven by an extension of the governor spindle, forces oil from a collecting tank at the base of the turbine, through a water-cooled coil, directly to the pipes distributing oil to the bearings, or to an elevated tank. An unloading valve returns oil to the suction tank if the delivery pressure is too high. Oil leaving the bearings passes through a filter to the suction tank.

479. Marine Steam Turbines. — The steam turbine is finding an increasing application on shipboard, principally because of the smaller space required as compared with reciprocating engines, the decreased expense for attendance and supplies, the decreased weight for the same power, and the greater cruising radius for the same weight of fuel. Its superior economy is accounted for, as in all steam turbines, by its ability to take advantage

of high-temperature steam without difficulty on account of lubrication, and of greater vacuum without the limitation of volume handled.

Turbines of the Kerr Economy type are made in sizes of 2500 to 3000 horsepower capacity for marine use and are used on merchant vessels of the Emergency Fleet Corporation. Since turbines cannot be reversed, a reversing element is carried on the shaft at the condenser end. When the turbine is running forward, these buckets are inactive and revolve in the exhaust connection surrounded by exhaust steam.

A 3000 horsepower **De Laval impulse marine turbine,** Fig. 409, is a multi-pressure stage turbine having eight wheels and eight sets of nozzles

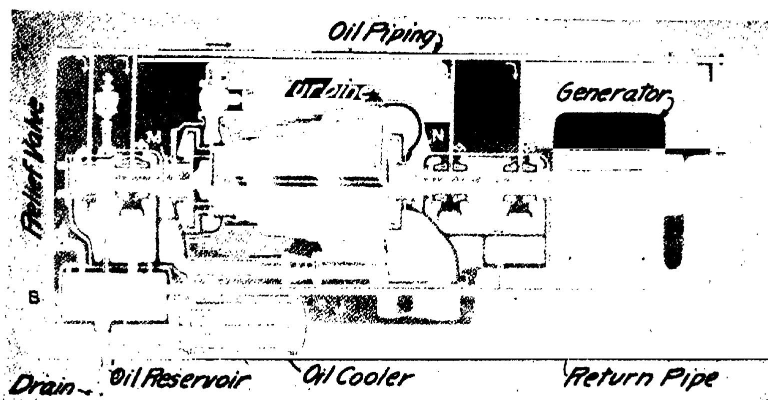


FIG. 408. — Forced Feed Oiling System for a Turbine.

for the **ahead turbine** and two wheels and two sets of nozzles for the **astern turbine.** These wheels are inclosed by a single casing split horizontally to allow easy access. The exhaust connection is to the upper half of the casing. The forward turbine has two velocity stages in the first pressure stage, and in each stage of the astern turbine. High- and low-pressure turbines with cross-compound cylinders are generally used.

A high-pressure **Westinghouse reaction marine turbine,** Fig. 410, shows the arrangement of the impulse element and the reaction element. The general construction is similar to the land reaction turbine except that a backing-turbine impulse wheel, having two velocity stages, is included in the casing. Balanced pistons are used to take the thrust resulting from the steam pressure, and a Kingsbury thrust bearing is used to take the thrust in either direction.

The economical speed of the steam turbine is too high for direct connection to the propeller shaft of vessels, where the speed is less than 17 knots* per hour. Since a large proportion of merchant vessels, as well as battleships at cruising speed, run at a lower speed, the successful application

* A knot is equal to a speed of 6080 ft. per hour.

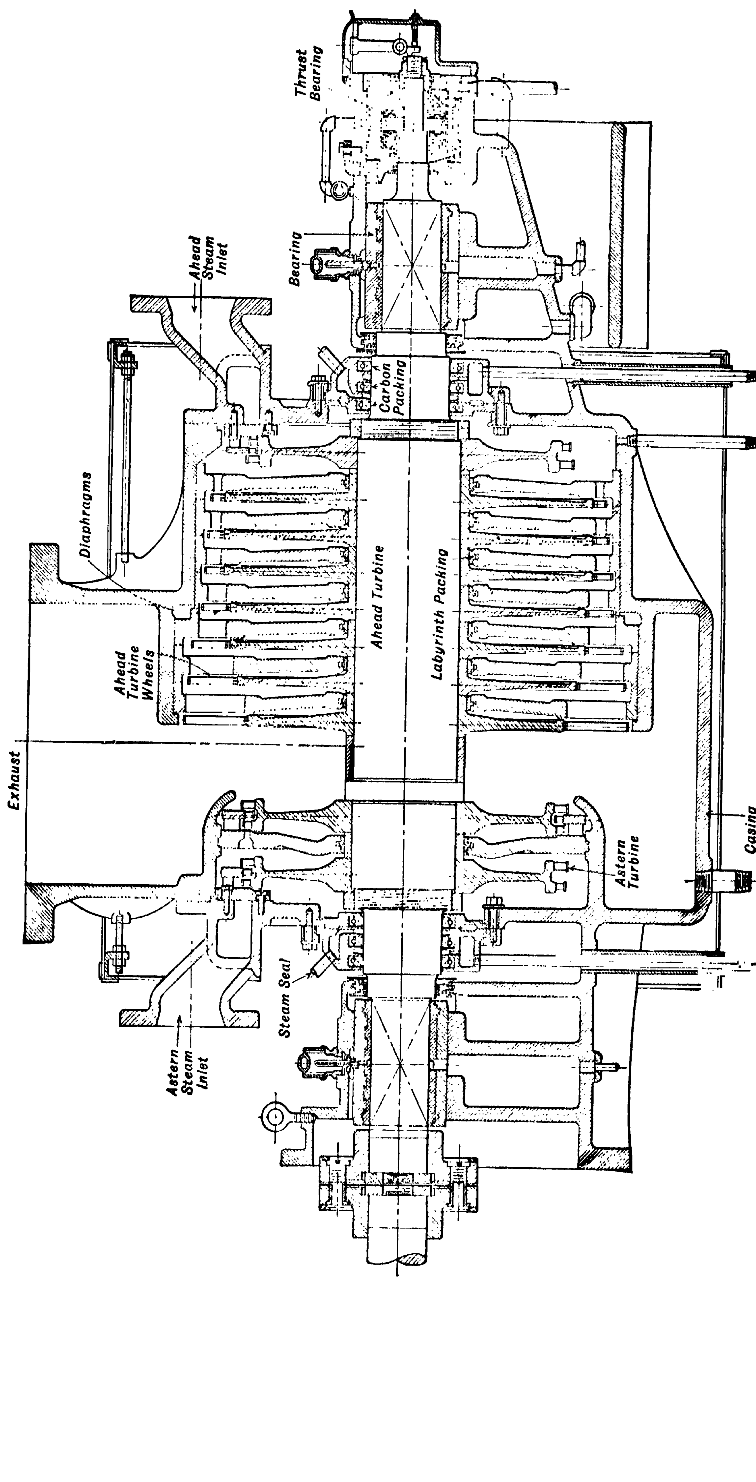


FIG. 409. — De Laval Impulse Marine Turbine.

of steam turbines to vessels requires some suitable means of speed reduction. As mentioned in Chapter I, the two methods used are (1) electrical transmission and (2) gearing.

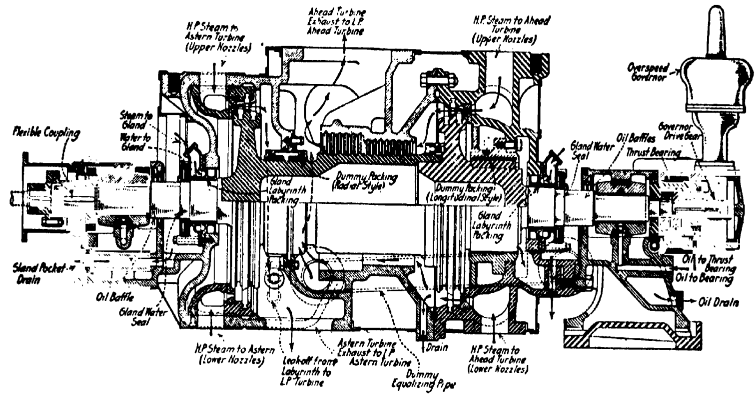


FIG. 410. — Westinghouse High Pressure Marine Reaction Turbine.

The electrical transmission is used on some recent battleships and employs **turbine-generator units** to furnish current for **electric motors** directly connected to propeller shafts. The power is generated by several

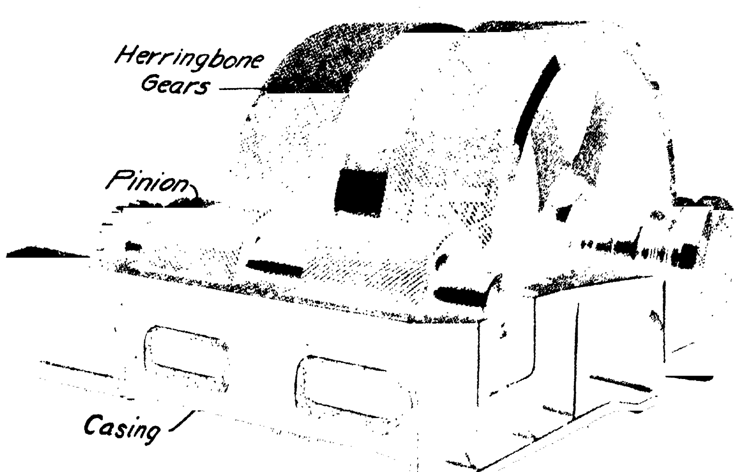


FIG. 411. — De Laval Two Pinion Reduction Gear.

turbine units, an arrangement which allows the propellers and turbines to be run at their economical speeds, and permits the adjustment of the load to give approximately full load on the turbines required for that speed.

480. Turbine Reduction Gearing. — Since the suitable speed of the propeller shaft is approximately 100 r.p.m., and the economical turbine speed varies from 5000 to 6000 r.p.m. for small turbines, to 1500 or 1800 r.p.m. for large turbines, double-reduction or single-reduction gears are required.

A two-pinion single-reduction gear, Fig. 411, consists of pinions driven by the turbine shafts meshing with a gear on the propeller shaft. Helical,

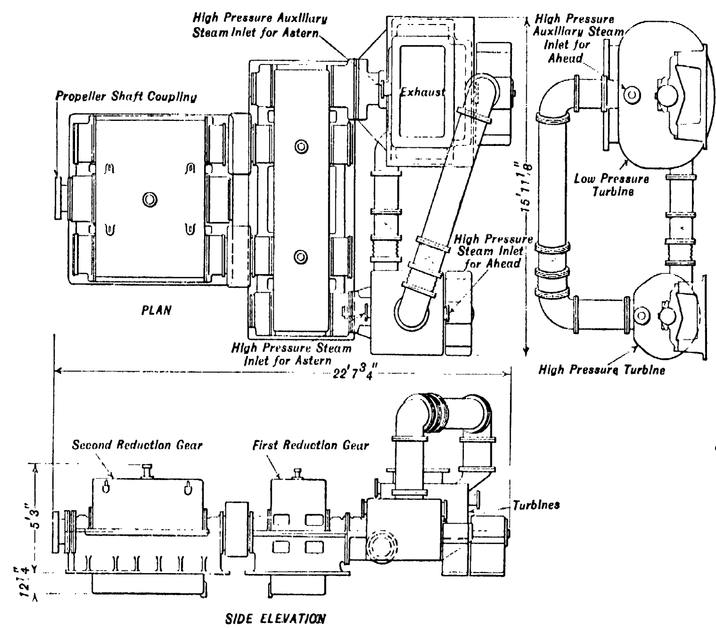


FIG. 412. — De Laval High and Low Pressure Marine Turbine with Double Reduction Gear; 8000 hp.; 90 R.P.M.

or twisted tooth, gears are used to insure continuous transmission of power. They are arranged in pairs, with the teeth inclined at equal angles to remove axial thrust. They run in oil supplied under pressure to the gear box, which holds the bearings of the pinion and gear and maintains proper alignment and spacing between the center of the gears. The location of a typical double-reduction gear, with reference to the turbine, is shown in Fig. 412.

The principal difficulty in the use of gearing, as applied to marine service, in order to prevent breaking the gears by concentrated load, lies in maintaining alignment under severe working conditions. Properly constructed gear cases will accomplish this, according to the experience of the De Laval Turbine Co. The bearings of the pinion shaft may be carried

by a floating frame, which is supported by hydraulic pistons, and thus maintains the alignment of pinion and gear even though the gear case be slightly distorted. This type of construction was designed by Melville and Macalpine for use in the United States Navy.

481. Losses in Steam Turbines. — The losses in steam turbines are of two kinds, namely, (1) heat or thermodynamic losses and (2) mechanical losses. The **thermodynamic losses** are the result of radiation, friction of the steam in the nozzles and buckets, loss by leakage past the buckets, stray motion and energy left in the steam, and loss to exhaust. The **mechanical losses** are caused by friction in bearings, and by rotation of the wheels in vapor-filled compartments. The distribution of the losses in a 200-kw. turbine-generator is given in Table 33, from " Steam Turbines," by Moyer

TABLE 33. — LOSSES IN A 200-KW. DE LAVAL TURBINE GENERATOR

Nature of Loss	Amount of Loss Per Cent
Nozzle losses	12
Radiation losses and leakage	1
Rotation losses caused by turbine wheel revolving in steam	4
Losses resulting from steam traveling over the blades	9
Bearing friction losses	1
Losses in speed-reduction gearing	2
Generator losses	4
Losses caused by residual kinetic energy in the steam passing to the condenser	8
Electrical output	59
Total	100

482. Economy of Steam Turbines. — The performance of steam turbines is stated as for steam engines, and is explained in Art. 399, page 399.

When using the water rate to compare the performance of turbines and reciprocating engines, the conditions of operation with regard to initial steam pressure, quality or superheat of steam, and final pressure should be the same. This condition is seldom attained, however, since there are special conditions for which both types of engines are designed in order to give the best results, and for this reason the performance is more commonly stated in heat consumption, as explained for engines in Art. 401, page 400.

In general it may be stated that the water rate of small turbines is higher than that of the same size of reciprocating steam engines, particularly if the operation is non-condensing. For turbines of medium sizes, up to 3000 kw., the water rate is slightly higher than that of piston engines of the best type. For larger sizes up to 6000 kw., the water rate of turbines

and engines is about the same, the difference being in favor of the turbine at 7500 kw. which is the size of the vertical-horizontal cross-compound condensing engines installed for the Interborough Rapid Transit Company at the 74th St. Station, New York. Piston engines above 3000 horsepower are seldom used in power stations.

A statement of the performance of steam turbines at their rated capacity is given in Table 34, taken from "Steam Power Plant Engineering," by GEBHARDT.

TABLE 34. — PERFORMANCE OF MODERN STEAM TURBINES AT RATED CAPACITY

Make of Turbine	Rated Capacity	Operating Conditions					
		R.p.m.	Initial Pressure Lb. Abs.	Back Pressure In. Mercury	Superheat ° F.	Lb. of Steam per kw.-hr.	Rankine Cycle Ratio
Westinghouse	500 kw.	3600	165	2.0	0	19.8	53.0
	1,000 kw.	3600	165	2.0	0	18.1	58.0
	5,000 kw.	1800	165	2.0	0	16.0	65.6
	15,000 kw.	1800	215	1.5	125	12.6	71.0
	30,000 kw.	1200	235	1.0	200	10.65	75.0
	45,000 kw.	1200	215	1.0	200	10.65	76.0
Curtis	500 kw.	3600	215	2.0	0	18.5	54.0
	1,000 kw.	3600	215	2.0	0	17.5	57.1
	5,000 kw.	1800	215	2.0	125	14.3	64.6
	15,000 kw.	1800	215	2.0	125	12.5	74.0
	30,000 kw.	1200	215	2.0	125	12.2	75.8
	45,000 kw.	1200	215	2.0	125	11.9	77.6
Kerr	25 hp.	3600	165	atmos.	0	43.0	31.0
	50 hp.	3600	165	atmos.	0	38.0	34.2
	100 hp.	3600	165	atmos.	0	32.0	40.6
	500 hp.	3600	165	atmos.	0	27.0	48.1
	1,000 hp.	3600	165	atmos.	0	24.75	57.5
	1,500 hp.	3600	165	atmos.	0	24.0	59.1

NOTE. — Rankine cycle ratio based on electrical horsepower for Westinghouse and Curtis, and on developed horsepower for Kerr.

483. Influence of Vacuum, Superheat, and Initial Pressure upon Water Rate. — The effect of increased vacuum is to decrease the water rate by approximately 5 per cent between 27 and 28 inches, and nearly 10 per cent between 28 and 29 inches vacuum. In modern stations, a vacuum of $29\frac{1}{2}$ inches is quite common. A comparison of the water rate of several sizes of turbines for condensing and non-condensing operation is given in Table 35, taken from "Steam Turbines," by MOYER.

An increase in pressure decreases the water rate, the decrease being 1 per cent for 10 pounds change in pressure. The present tendency is toward higher steam pressures, and the upper limit seems to be set by the mechanical difficulties caused by high temperature.

The water rate decreases 1 per cent for each increase of 12 deg. fahr. in the superheat.

484. Testing of Steam Turbines. — In general, the method used to test steam turbines is the same as that used for steam engines, with the modifications which the difference in method of operation requires. When guarantee tests of turbines are made at other conditions of pressure, quality, superheat, and vacuum than those stated, it is customary to correct the test to the guarantee conditions, provided the variation is not too great, by using curves furnished by the manufactures, which show the correction to be made in each factor. Consult "Steam Turbines," by MOYER, for a discussion of the method to be used in making these corrections. For detailed information on testing of steam turbines consult A. S. M. E. POWER TEST CODE.

TABLE 35. — COMPARISON OF WATER RATES FOR 500-KW. TURBINE OPERATING CONDENSING AND NON-CONDENSING AT 3600 R.P.M.

Brake Horsepower	Steam Pressure Lb. per sq. in., Gage	Superheat deg. fahr.	Vacuum, inches	Pounds per b.hp.-hr.
383.5	152.6	0.2	28.2	14.15
755.6	149.2	1.2	27.8	13.28
1121.9	148.8	5.1	26.5	14.32
385.6	148.2	2.7	0.8	24.94
766.8	147.3	2.6	0.8	22.10
1144.4	126.1	11.4	0.8	24.36

REFERENCES

Mechanical Engineers Handbook, MARKS.
Steam Power Plant Engineering, GEBHARDT.
Applied Thermodynamics, ENNIS.
Steam Turbines, ROE.
Steam Turbines, MOYER.
Elements of Heat Power Engineering, HIRSHFELD AND BARNARD.

Publications of the following companies:
GENERAL ELECTRIC CO.,
DE LAVAL STEAM TURBINE CO.,
WESTINGHOUSE ELECTRIC AND MFG. CO.,
TERRY TURBINE CO.

REVIEW QUESTIONS AND PROBLEMS

1. In what fundamental respects does a steam turbine differ from a reciprocating steam engine?
2. State the essential difference between the impulse and reaction types of turbines.
3. What is meant by a stage in a turbine, and what are the two methods of staging used? What is the purpose of staging?

4. Describe the construction of the following turbines: (*a*) Simple type of De Laval, (*b*) Curtis horizontal, (*c*) Westinghouse reaction, (*d*) Terry.

5. What methods are used to govern steam turbines? Describe the method of governing (*a*) De Laval, (*b*) Curtis, (*c*) Westinghouse, turbines.

6. What types of gland packing are used, and what is the purpose of such packing?

7. Describe the forced oiling system, as applied to a turbine.

8. The steam pressure on a turbine nozzle is 150 lb. per sq. in. abs., and the pressure at discharge 14.7 lb. per sq. in. Find the theoretical velocity of the issuing jet, assuming 10 per cent friction loss and dry steam.

9. What would be the impulse force produced by 3.5 pounds of steam discharged per second from the nozzle in Problem 8?

10. Explain the meaning of (*a*) low pressure, (*b*) bleeder, and (*c*) mixed pressure as applied to turbines.

11. What is a double-flow turbine, and why are turbines made with this construction?

12. Describe the construction of a marine impulse turbine.

13. Name the losses occurring in a steam turbine.

14. What effect have the following upon the water rate of a turbine: (*a*) increase in steam pressure, (*b*) decrease in superheat, (*c*) increase in vacuum?

15. The performance of a certain line of turbines, in 1899, was as follows: load, 300 kw.; steam pressure, 125 lb. gage; superheat, 0 deg. fahr.; vacuum, 27 in. mercury; steam per kw. per hr., 22 lb. In 1920 this type of turbine gave: load, 26,505 kw.; steam pressure, 233.1 lb. gage; superheat, 124.3 deg. fahr.; vacuum, 28.85 in. mercury; steam per kw. per hr., 11.27 lb. The barometer in both cases was 30.05 in. mercury. Find the efficiency, or Rankine Cycle, ratio for the two turbines.

CHAPTER XXIV

STEAM AND POWER DRIVEN PUMPS

486. Foreword. — Pumps are used in power plants for the following purposes: to pump water into boilers; to circulate cooling water in, and to remove air and vapors from, condensers; to pump lubricating oil; and to supply water for fire protection.

Until recently, the piston pump, driven by a small uneconomical steam engine, has been most commonly used. This type of pump is reliable and gives satisfactory service. The exhaust steam is available for heating the boiler feedwater, and the loss resulting from the poor economy of the driving engine is, in a measure, overcome.

The perfecting of the steam turbine and the centrifugal air compressor also brought about the perfecting of the centrifugal pump. This type of pump is now used extensively to feed boilers and to circulate cooling water in condensers. It has few operating parts, requires only a small space, can be operated at high speeds, is economical in use of steam, and requires a minimum of attention.

Air pumps are described in Chapter XXV.

487. Classification of Pumps. — Pumps used for water, in modern power plant practice, may be classified according to the principle of operation as:

1. Piston
2. Centrifugal
3. Rotary
4. Jet
5. Direct pressure

488. Piston and Plunger Pump. — In this type of pump, motion and pressure are imparted to the fluid by direct contact. The action is positive, and a definite amount of fluid is pumped per stroke for a given pressure and velocity.

Piston pumps may be subdivided as follows:

1. Direct-acting, steam-driven	Simplex Duplex
2. Flywheel	
3. Power-driven, electrically, or by belt	Simplex Duplex Triplex

489. Direct-acting Steam-driven Piston Pump. — *A steam pump that has no flywheel is known as a direct-acting pump.* Such pumps have a small number of working parts, require little attention, and are generally reliable. For general power plant service, they are used more extensively than any other type As built, they may have a single steam cylinder and a single water cylinder, in which case they are **simplex;** or they may have two steam and two water cylinders, in which case they are **duplex.** The size of this type of pump is stated by giving first the diameter of the steam cylinder, then the diameter of the water cylinder, and finally the length of the stroke, all in inches; thus, 8 in. × 6 in. × 10 in.

490. Simplex Steam-driven Pump. — The simplex pump has a single steam piston and a single water piston, connected to the same piston rod. The cylinders are generally arranged to have the pistons at opposite ends

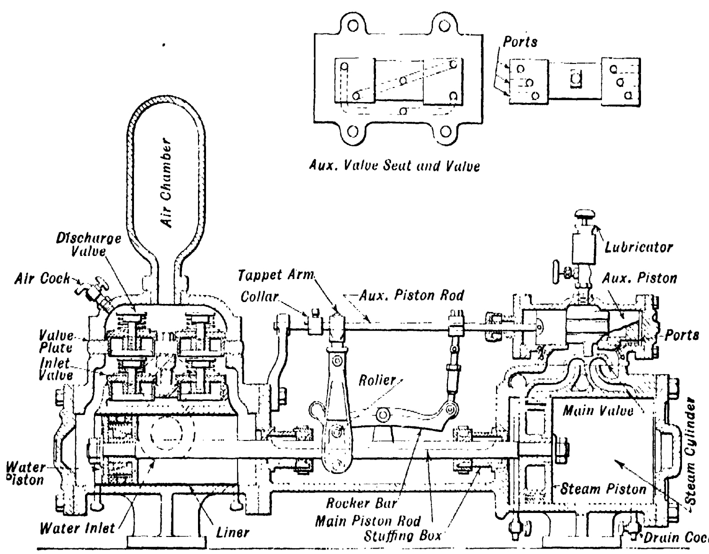


FIG. 413. — Knowles Simplex Direct-acting Steam Driven Pump.

of the piston rod, with a connecting casting between. Reversal of the stroke is obtained by using a **steam-thrown steam valve,** of which there are many types, and which takes the place of a flywheel. The principle of operation is the same in all types, and will be illustrated by the description of two typical pumps.

491. Knowles Steam-driven Simplex Pump. — This pump is shown in Fig. 413, with the essential parts named. The steam cylinder and valve chest of the pump are practically the same as those described under the steam engine, with the addition of a special control of the slide valve.

The operation of the steam end of the pump is as follows: The **auxiliary piston** is attached to and drives the **main steam valve** back and forth, upon its flat seat; steam is thus admitted to the cylinder, back of the main steam piston, and the pump operated. The main steam valve is a plain slide valve having the form of the letter *B*. It does not have any lap, and hence steam at full pressure is used for the entire length of the stroke.

Movement of the auxiliary piston is accomplished by giving it a slight rotary motion at the proper time. This rotation, relative to the steam chest, brings the small steam ports, located on the under side of the auxiliary piston, to coincide with the proper openings cut in the walls of the steam chest. When the ports coincide, steam from the steam chest is admitted to that end, back of the auxiliary piston, and this piston, together with the main steam valve, is moved toward the opposite end. This movement of the main valve connects one side of the main piston to admission and the other to exhaust. To prevent the auxiliary piston from striking the steam chest head, a port is so located that it is uncovered by the auxiliary piston after it has traveled a certain distance, and steam is admitted to cushion the piston.

Rotation of the auxiliary piston is produced by a rocker bar, connected to the side of the frame and attached to the auxiliary piston rod. The rocker bar is moved by a **roller** located on the **tappet arm,** which is fastened to and moves with the main piston rod. As the main piston nears the end of its stroke, the roller comes in contact with the rocker bar, raising or lowering it and partially rotating the auxiliary piston. The upper end of the tappet bar ordinarily does not come in contact with the **collars** on the auxiliary valve rod. If the steam pressure for some reason should fail to move the auxiliary piston, the tappet arm strikes the collars and moves the auxiliary piston mechanically.

At the position of the steam end shown in Fig. 413, steam is about to be admitted to the right of the auxiliary piston. When this occurs the main valve will be moved to the left; steam will be admitted to the left-hand side of the main piston, its motion will be reversed, and the space to the right of the main piston will be connected to exhaust.

The operation of the water end is as follows: At the position of the water end shown, water has just ceased to flow through the water passage at the head end and upward past the discharge valves located in a removable **valve plate** above the inlet valves. During the discharge from the head end of the pump, the crank end discharge valves are held closed by the pressure of the water in the discharge chamber and the inlet valves are opened, by the unbalanced pressure produced by the movement of the piston to the left, and water enters to fill the space behind the moving piston. When the direction of movement of the piston is reversed, by the action of the valves in the steam cylinder, the crank-end inlet valves are closed by the pressure

of the discharge water, which flows through the crank end discharge valves into the outlet pipe, shown dotted in the figure. During discharge at the crank end, the head-end discharge valves are closed and the inlet valves are forced open by the pressure on the water in the suction chamber, and water again enters to fill the space behind the piston. The discharge pressure for a pump of this type depends upon the relative areas of the steam and water pistons.

An **air chamber** is attached to the discharge, to maintain a uniform pressure in the discharge line and to produce a nearly uniform rate of flow by storing water when there is an excess and delivering water when there is a deficiency.

The **water** or **pump valve** used on this type of pump, and shown in Fig. 414, consists of a rubber-composition disk supported by a brass casting and held closed by a coil spring. The **removable seat** is made of brass to

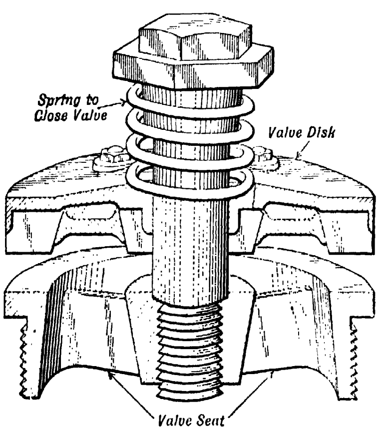


FIG. 414. — Sectional View of Hill Pump Valve.

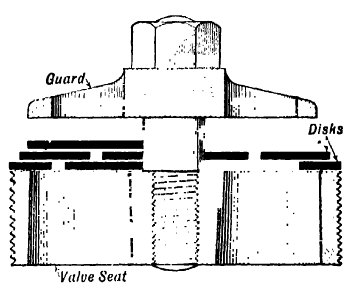


FIG. 415. — Multi-disk Type Pump Valve.

prevent corrosion and rusting. Several small valves are used instead of one large one, because they give more satisfactory service. The discharge valves are located in a removable **valve plate.**

A recent type of pump valve is shown in Fig. 415. It consists of a number of superimposed metallic disks, each being free to move on a central stud. The bottom disks contain a number of holes, so placed that when all the disks are closed no passageway is formed. The disks rise from their seats upon each other, independently or together, to permit water and air to pass.

491. Cameron Steam-driven Simplex Pump. — The water end of the Cameron pump, Fig. 416, has the same general construction as the

Knowles pump, but the operation of the steam end is different, in that the main piston is reversed by using two plain tappet valves which control exhaust passages to the auxiliary piston. All the operating parts are enclosed.

The main slide valve is moved by the auxiliary piston, which is hollow at each end and filled with steam from the main steam chest. Steam passes through a small hole in each end of the auxiliary piston, and fills

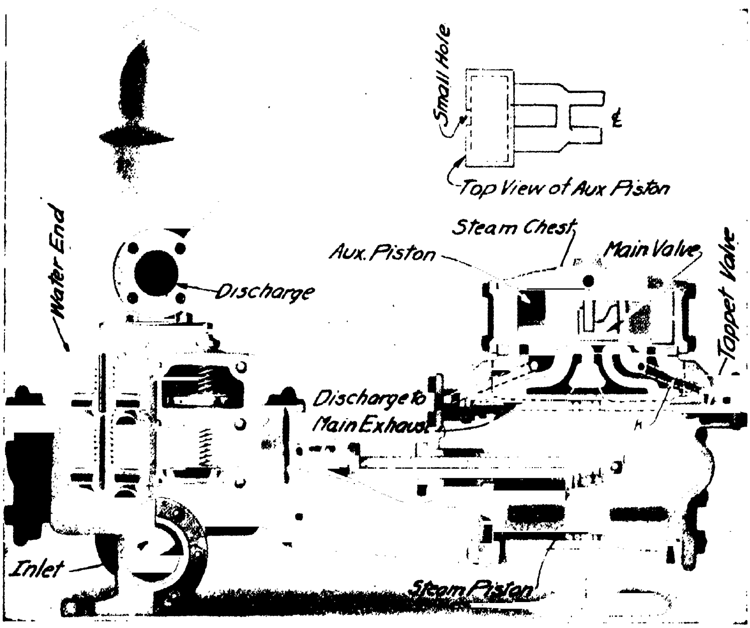


FIG. 416. — Cameron Steam Driven Simplex Pump.

the space between the ends of the piston and the heads of the steam chest in which it moves The pressure acting on each end of the auxiliary piston is thus equal, and the piston is ordinarily balanced and motionless. When the main piston has traveled until it strikes the reversing tappet, at either end, steam is discharged through a small exhaust port, *E*, from that end of the piston valve, to the main exhaust passage. This causes an unbalanced pressure on the piston valve, which moves, carrying with it the main valve and thus reversing the piston of the pump. When the piston valve passes the port *E*, it encloses steam and forms a cushion. As soon as the main piston moves out of contact with the reversing tappets, they are closed by steam pressure conveyed directly from the steam chest through ports *K* shown by broken lines.

492. Direct-acting Duplex Steam-driven Pump. — A sectional view of a Deane Brothers duplex direct-acting steam pump is shown in Fig. 417.

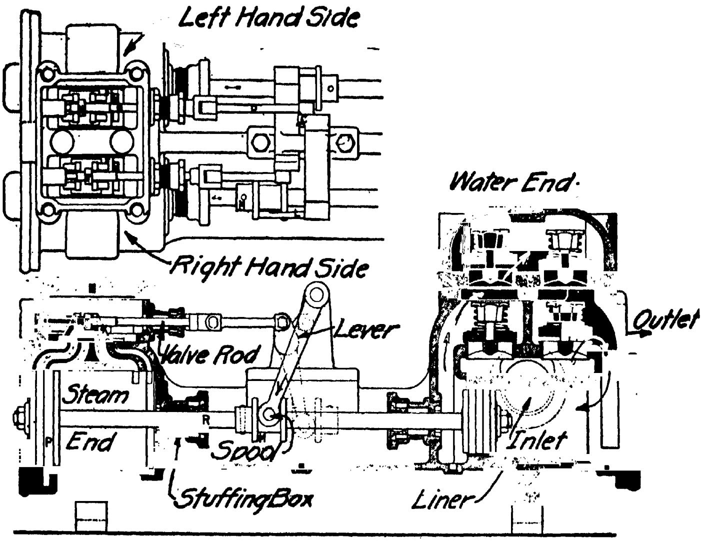


FIG. 417. — Duplex Steam Driven Pump.

This pump has two steam cylinders and two water cylinders set side by side, and is, in effect, made up of two pump cylinders in one casting.

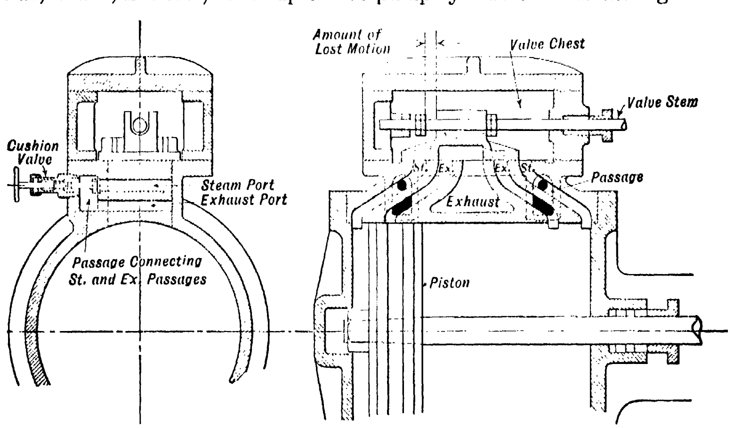


FIG. 418. — Steam Cylinder of Duplex Pump Showing Cushion Valves.

The steam valve of each side is operated by a connection to the piston rod of the opposite side, thus making the operation of the steam valves

positive. To a person standing at the steam end, the pump at the right is known as the **right-hand side** and the other as the **left-hand side.**

As one piston moves to the end of its stroke and is gradually brought to rest, it moves the slide valve of the opposite steam cylinder, admitting steam back of the piston, which is at rest, and causing it to move forward to the opposite end of its stroke.

The valves are plain slide valves, without lap or lead. Lost motion is allowed between the valve and its operating mechanism, which causes the piston to pause at the completion of the stroke. The pistons are in motion only about five-eighths of the time. The length of stroke is varied by changing the lost motion at the valve. The stroke is never constant, but changes with the discharge pressure.

To prevent the piston striking the cylinder head at each end of the stroke, the steam cylinder is made with five ports, Fig. 418, two outside end ports for admission of steam and three inner ones for the exhaust. When the piston closes the exhaust port in the steam cylinder, steam is trapped between the piston and the cylinder head, thus cushioning the piston. The amount of this cushion is ordinarily controlled by a small hand-operated valve, located at the side of the steam chest which opens or closes a passage connecting the outer steam and exhaust passages. With this valve nearly closed, the cushion steam can escape slowly by way of the steam passage through the cushion valve, to the exhaust passage. This should be the condition when running rapidly and at light loads. With the cushion valves wide open there will be only a slight cushioning; and the length of the stroke will be a maximum.

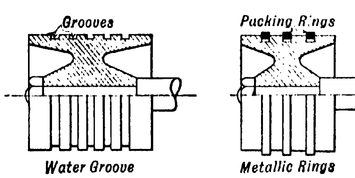
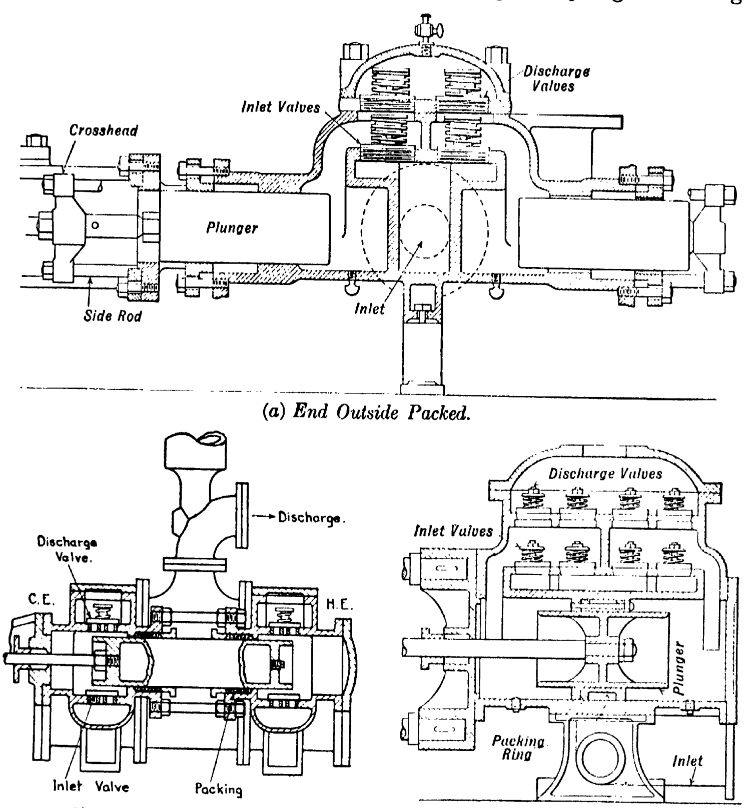


FIG. 419. — Types of Pump Water Pistons.

The duplex steam-driven pump is often made with the steam end tandem compound, the valves for both cylinders being connected to the same valve rod and operated in the same manner as the valves on an ordinary duplex pump. With this construction the steam is used expansively, since the low-pressure cylinder has a volume several times that of the high-pressure cylinder and steam is used in each cylinder for the entire stroke. A pipe connects the exhaust opening of the high-pressure cylinder to the steam chest of the low-pressure cylinder.

493. Types of Pump Water Ends. — Most of the simplex pumps of the less efficient types have pistons, which are constructed as shown in Fig. 413, and have a space into which durable water packing, made of **layer canvas,** is placed. The cylinder is generally fitted with a **brass liner** to prevent rust, and the consequent rapid wear of the packing; and when thus made, a wide piston with grooves, Fig. 419, made on its face is often used. The

fit between the piston and liner and the effect of the grooves are relied upon to prevent leakage. A plunger, Fig. 420c, is often substituted for the piston, and leakage is prevented by a packing held in place by a gland through



(*a*) *End Outside Packed.*

(*b*) *Center Outside Packed.* (*c*) *Inside Packed.*

FIG. 420. — Types of Plunger Pumps.

which the plunger slides. In the above types the packing is inside the cylinder, where leakage cannot be easily detected, and to renew the packing the cylinder head must be removed.

To make leakage past the plunger visible, outside packed plunger pumps are used. The leakage is then easily prevented by adjusting the gland against the packing. In Fig. 420*b*, only one plunger is used with the packing at the inner end of the cylinder; while in Fig. 420*a* two plungers, connected by a yoke so that they move together, are used, and the packing is placed at each outside end of the cylinder. In this latter pump a partition separates the compartments in which the plungers work.

494. Power-driven Pumps. — Piston pumps driven by belting, chains or gearing are classed as power-driven pumps. The source of the power may be steam, gas, or electricity, and the speed is ordinarily constant. Pumps having only one plunger are called **simplex;** those with two plungers, **duplex;** and those with three plungers, **triplex.** They are used for the kind

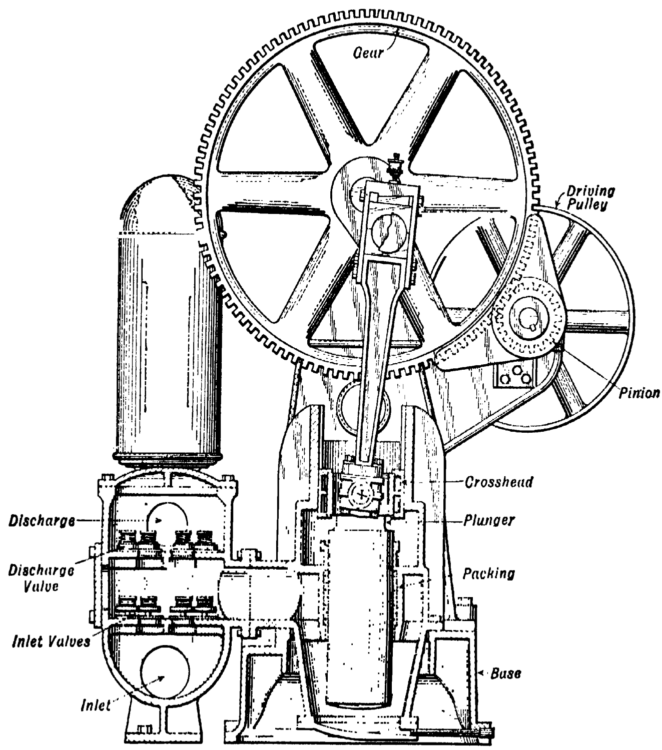


FIG. 421. — Goulds Single Acting Triplex Plunger Pump.

of service formerly performed by the direct-acting steam pump. When driven by electric motors and running at high speeds, the **overall efficiency** of this type of pump may be as high as 83 per cent.

Figure 421 shows a typical triplex power-driven pump, in which each plunger is driven by means of a crank and connecting rod. The upper end of each plunger is shaped to form a crosshead, and a gland with packing prevents leakage at the point where the plunger enters the pump base. The base is made in compartments, so that the operation is the same as for the water end of any pump, and an air chamber is provided on the discharge side.

495. Suction Air Chambers. — On long suction lines, the shock caused by stopping a large body of moving water is taken up by using **suction air chambers,** Fig. 422, which should be near the pump and in direct line with the current. By this means the pump is made smoother running, less noisy, and more efficient, because the head of water on the pump is fully utilized. The life of the pump is also increased and repair bills are saved.

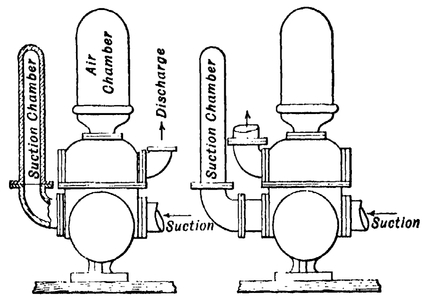


Fig. 422. — Arrangement of Suction Air Chambers for Pumps.

496. Pump Governor. — Steam-driven pumps are generally equipped with a pump governor which maintains a nearly constant pressure in the discharge pipe of the pump, irrespective of the quantity of water flowing. It does this by controlling the speed of the pump. A typical governor, Fig. 423, consists of a pressure-reducing valve located in the steam supply-pipe of the pump and moved by slight variations in the discharge water pressure which acts upon the top of a small piston attached to the double balanced valve in the steam line. A spring attached to the valve spindle normally holds the valve open, and a small regulating wheel on the valve spindle of the governor can be adjusted to change the amount of discharge pressure. A change in this pressure, acting on a small piston at the upper end of the valve spindle, causes a difference in pressure between that caused by the water and that resulting from the spring. The balanced valve then adjusts itself to this differential pressure and regulates the discharge pressure by changing the amount of steam admitted to the pump. This changes the speed and gives the desired discharge pressure.

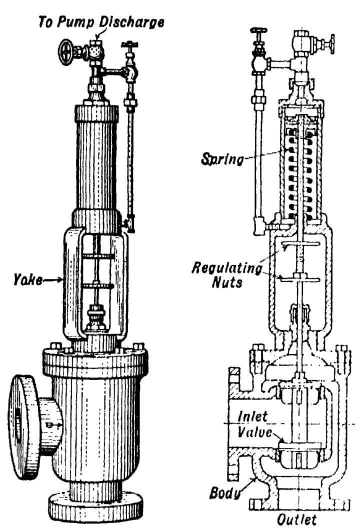


Fig. 423. — Fisher Spring Controlled Pump Governor.

Power-driven pumps generally run continuously, at a constant speed,

and deliver sufficient water to supply the maximum demand. To allow for variation in the demand, a by-pass connection, containing a gate valve, a relief valve, and a check valve, is provided. The amount of feed may be regulated by hand, or by means of a pressure regulator similar to the pump governor. The surplus water returns to the source of supply.

497. Centrifugal Pumps. — This type of pump is, to a considerable extent, supplanting the piston pump. It is particularly adapted for low

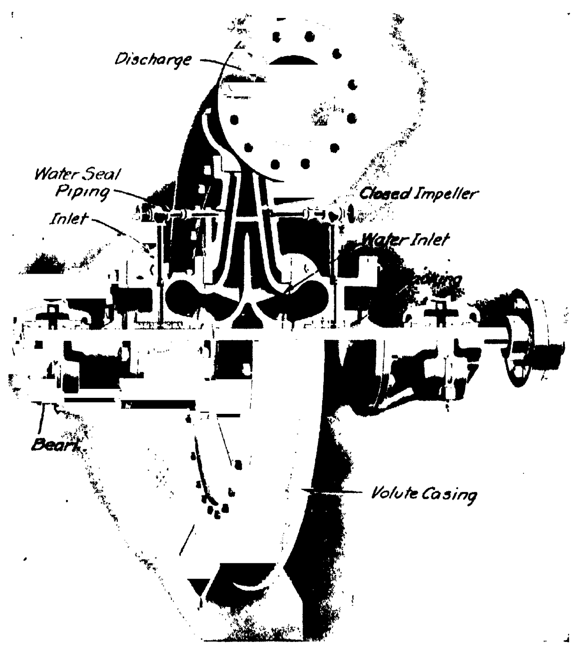


FIG. 424. — Wheeler Closed Impeller Centrifugal Pump.

heads and large volumes; by using stages, however, high pressure may be obtained. The average mechanical efficiency of centrifugal pumps varies from 70 to 82 per cent, according to the load. The cost is low, and this offsets the higher mechanical efficiency of high-grade pumping engines. In power plants, these pumps are used to circulate water through condensers, to pump boiler feedwater, and for fire duty. They are generally direct-driven by a steam turbine or an electric motor. The pump shown in Fig. 424 may be driven electrically or by a turbine. When the impeller revolves in a right-handed direction, viewed from the driving side, the pump is known as a **right-hand centrifugal pump.** When revolving in the opposite direction, the pump is **left-hand.**

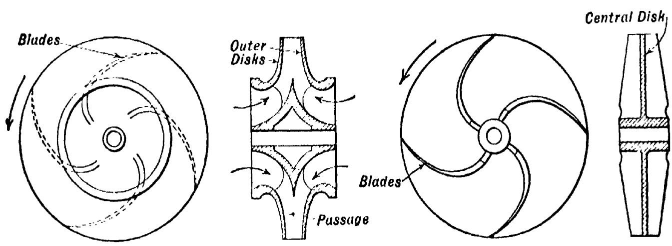
A centrifugal pump has the following essential parts:

1. A **rotary impeller** to increase the velocity and pressure of the water, which enters at the center of the impeller and is discharged at its circumference with increased velocity and slightly increased pressure.

2. A cast-iron **casing** having inlet and outlet passages. The casing guides the water from the circumference of the impeller to the discharge outlet, and is often shaped to convert the velocity, or kinetic-energy, of the water to pressure, or potential energy.

3. A **shaft,** which supports the impeller; and bearings in which the shaft revolves.

The impeller may be of the open or the closed type, Fig. 425. The **open impeller** consists of a circular central disk partition to which are attached blades having an **involute form.** This type of impeller does not fit closely in the casing within which it revolves, and consequently the guidance of the



(*a*) *Closed Type, Double Suction* (*b*) *Open Type.*

Fig. 425. — Types of Impellers for Centrifugal Pumps.

water and the flow lines are poor. In addition, a large amount of water slips between the revolving impeller blades and the casing wall, thereby reducing the efficiency.

The **closed impeller** has two outer disks separated by blades, similar in form to the open-type blades, which form passages for the water. The water enters the passages at the center, and is guided to the outlet. The side walls of the impeller prevent water from slipping past, and thus the efficiency is improved.

With either of these types of impellers, the water may enter from one or both sides. The **double-suction** type is preferable, as it minimizes the end-thrust on the shaft, produced by the unbalanced water pressure, which exists when water enters from one side only. The **single-suction** impeller must be balanced, to overcome the end-thrust.

Centrifugal pumps are generally classified, according to the construction of the casing, as:

1. Volute.
2. Turbine.

The **volute pump** does not use a diffusing, or guide, vane surrounding the impeller, but has a **spiral,** or **volute, casing** to guide the water, as it

leaves the impeller on its way to the discharge pipe. The shape of the volute is such that it converts the velocity of the water leaving the impeller

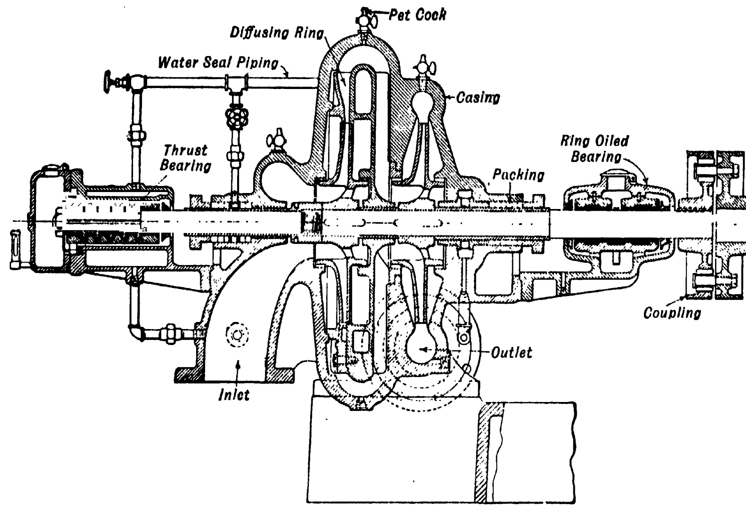


Fig. 426. — Cameron Turbine-type Centrifugal Pump — 2 stage.

into pressure. In this type of pump, the water leaving the impeller is thrown across the stream of water in the casing, and the flow is disturbed.

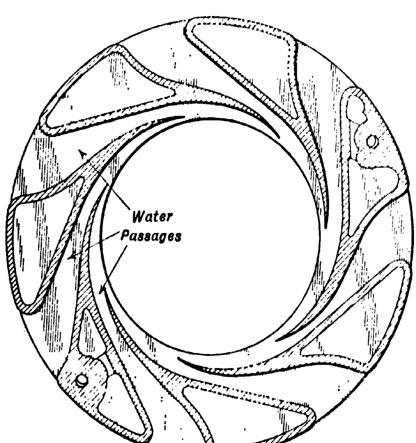


Fig. 427. — Diffusing Ring for Turbine-type Centrifugal Pump.

The **turbine pump**, Fig. 426, uses a circular diffusing ring, Fig. 427, containing ribs which form passages for the discharge of water. This ring surrounds the impeller, and the passages are shaped to convert the velocity of the water leaving the impeller into pressure. It also directs the water tangentially into the casing, which may be concentric with the impeller and of uniform cross section, or it may be volute. With this type of construction, the efficiency may be as high as 82 per cent. The mechanical efficiency attained depends upon the shape of the impeller and casing, and upon the number of stages used.

498. Compound Centrifugal Pump. — When it is desired to increase the discharge pressure above 100 pounds per square inch, a compound cen-

trifugal pump, Fig. 428, is used. Several impellers are mounted on a shaft, and revolve in separate compartments, or stages, of the main casing.

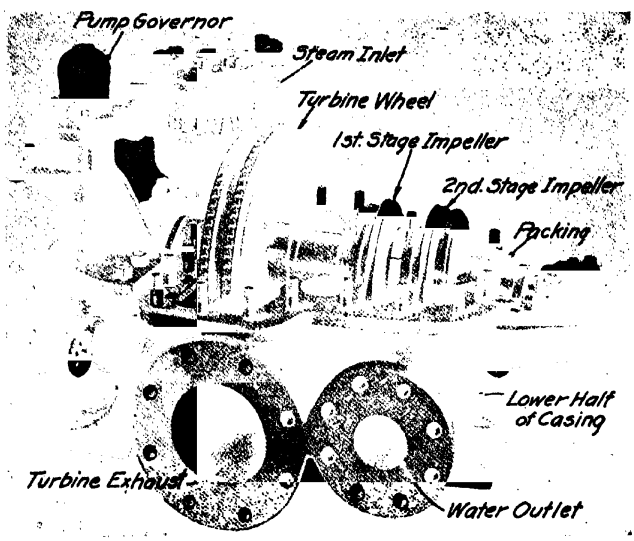


FIG. 428. — De Laval Turbine-driven Compound Centrifugal Boiler-feed Pump.

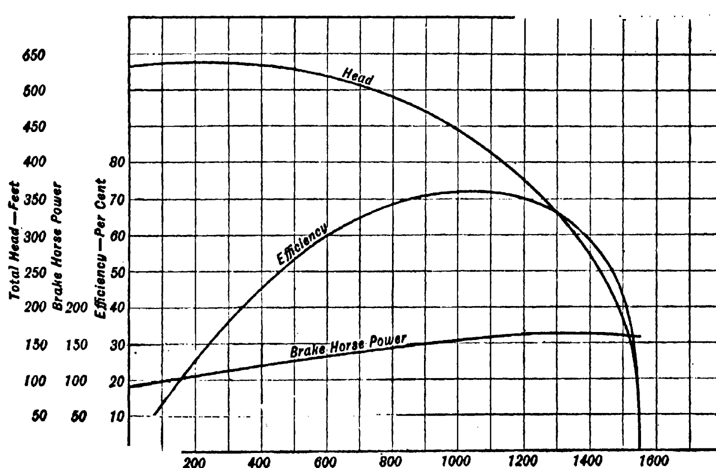


FIG. 429. — Typical Characteristics of a Goulds No. 6, 3-stage Centrifugal Pump, R.P.M. 1740.

The discharge from each stage is delivered to the suction of the next higher stage, and the final discharge pressure depends upon the number of stages. Closed single-inlet impellers are generally used.

499. Performance of a Centrifugal Pump. — Characteristic curves of a typical centrifugal pump, given in Fig. 429, show that the efficiency varies with the capacity. *The capacity is generally stated at the maximum efficiency and to obtain the best efficiency this pump should be operated at this capacity.*

500. Rotary Pump. — This pump, Fig. 430, is used for the same purposes as the centrifugal pump; the small sizes are also used to pump oil for lubricating purposes. It can be operated at a low speed of rotation, the pressure attained is moderate, the volume discharged is large, and the space required is small. At high speeds, however, this type of pump is noisy. It consists of an accurately machined casing, having an inlet and outlet opening. Within the casing are two rotors attached to separate shafts, which are connected by gearing and to one of which the power is applied. Each rotor, or impeller, is accurately machined to run with a small clearance within the casing, and to mesh with the other rotor. Water enters at the bottom, is enclosed within the pocket formed between the rotors, and discharged at the top. The volume delivered varying with the speed at which the pump is operated. There are many forms of rotors which operate like those described.

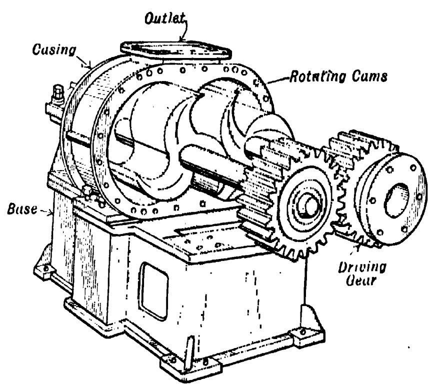


FIG. 430. — View of Goulds Rotary Pump, Front Cover removed.

501. Jet Pump. — This type of pump is used mainly for feeding water into a boiler, where its high thermal efficiency justifies its use, because nearly all the heat used to operate the pump is returned to the boiler as warm feedwater. Multi-jet pumps, for removing air from steam condensers, are giving great promise of high efficiency, Art. 532, page 540. For general pumping service the mechanical efficiency of a jet pump is low.

The **injector** is a typical jet pump. It is efficient, convenient, cheap, compact, has no moving parts, delivers warm water into a boiler without preheating, and has no exhaust to be disposed of. It is universally used on locomotive boilers, and in connection with feed pumps as a reserve on large land boilers. It will not handle hot water. The essential parts of the injector are illustrated in Fig. 431, steam from the boiler enters at the top and flows to the atmosphere through the **steam tube, combining tube** and **overflow.** *The steam tube is so shaped that it converts the pressure energy of the steam into velocity energy.* As a result of the high velocity attained

in the steam tube, the air is partially exhausted from the inlet pipe, thereby causing the water to rise until it comes in contact with the steam at the entrance to the combining tube. The steam, coming in contact with the water, is condensed and imparts considerable momentum to the water. The condensing of the steam reduces its volume and serves to maintain the vacuum. The shape of the combining tube is such that the velocity of the moving mass of water is increased sufficiently to carry it across the opening to the **delivery tube,** in which the *velocity of the water is partially converted into pressure.* The water lifts the check valve, because of this pressure and momentum, and enters the boiler against boiler pressure. The final discharge pressure depends upon the shape of the discharge nozzle. When the injector is operating properly, water will not show at the overflow.

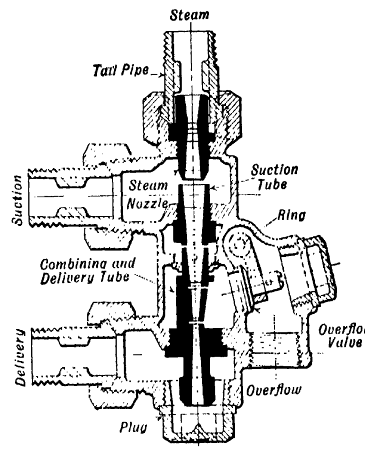


FIG. 431. — Interior View Metropolitan Automatic Injector.

Injectors are either **hand-starting** or **automatic,** and either **single-** or **double-tube.** The automatic injector will resume its flow, after an interruption, without any attention from the operator.

502. Automatic Injector. — A single-tube automatic injector is shown in Fig. 431 with the operating parts marked, so as to require no description. The operation is similar to the general description in the previous article, with the exception that the combining tube is surrounded by a loose ring, which normally remains closed when in operation. Should the discharge be interrupted, this washer opens and passes water to the overflow until the vacuum is again established, when it closes and the injector again delivers water. It is started by opening the steam valve in the steam pipe connecting the injector to the boiler.

503. Double-tube Injector. — The Hancock double-tube injector, Fig. 432, illustrates this type, which consists of a double set of nozzles, called the **lifting set** and the **forcing set.** The lifting set is the lower set and consists of a steam nozzle and a combining tube. The forcing set of tubes consists of a steam nozzle, combining, and delivery tube. The valve which controls the steam flow to the forcing set is attached to a lever, to which the overflow valve is also attached. When starting, the lever is only partly open and this holds the overflow open. When started, and with

the lever wide open, the overflow is closed This type of overflow is called a **closed overflow.**

To operate, the *hand-operated regulating steam valve, which controls the amount of steam used, is partly opened.* This admits steam to the lifting steam nozzle, then through the lifting combining tube into the overflow chamber, to the atmosphere. The rapid passage of steam creates a vacuum

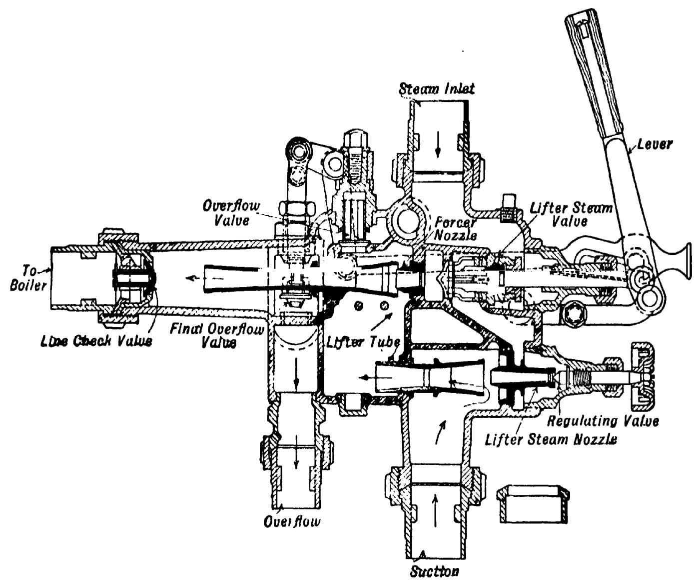


FIG. 432. — Hancock Double-tube Injector.

in the suction chamber, whereupon water flows in, mingles with and condenses the steam. When water appears at the overflow, the lever is pulled back to the stop and the forcing nozzle put into operation. The water discharged from the lifting combining tube is delivered to the forcing combining tube, where the forcing jet meets and mingles with it and delivers the water into the boiler, as in the injector previously described. *The use of a double-tube injector increases the delivery pressure and permits varying the capacity by regulating the steam inlet valve to the lifting steam nozzle.* This injector must be re-started by hand when the flow is interrupted.

The **capacity of injectors** is commonly stated in cubic feet or gallons of feedwater passing through the delivery tube per hour. The operation of an injector is affected by the delivery pressure; the temperature of the feedwater; and the **lift,** or the distance the injector is above the supply.

504. Terms Used in Connection with a Pump. — Capacity of a pump is generally stated as the number of gallons per minute the pump will deliver.

For piston pumps, manufacturers generally state the capacity as the piston displacement per minute.

Slip is the difference between the quantity of water actually delivered and the piston displacement, both expressed in cubic feet per minute. It varies from 5 to 20 per cent, and is greater at small outputs. The amount of slip depends upon the tightness of the valves, the action of the valve springs, the fit of the piston and the general condition of the pump.

The **horsepower** of a pump is generally called **water horsepower,** and equals the weight of water in pounds actually delivered per minute multiplied by the distance through which the water is lifted divided by 33,000, or

$$\textit{Water horsepower} = \frac{Wh}{33{,}000} \quad \ldots\ldots\ldots \quad (104)$$

in which W = pounds of water delivered per minute.
h = head in feet = pressure in pounds per square inch $\times$ 144 $\div$ density of water pumped.

Example 53. — During the test of a 432,000-gallon Fairbanks Morse compound duplex pump, 133,250 lb. of water were delivered per hour against a pressure of 121.8 lb. per sq. in. gage; temperature of water, 70 deg. fahr. Find the water horsepower, considering the discharge pressure as the only resistance to be overcome.

Solution. — Using Equation (104),

$$\text{Water horsepower} = \frac{Wh}{33{,}000} = \frac{2221 \times 282}{33{,}000} = 18.9$$

$$W = \frac{133{,}250}{60} = 2221 \text{ lb. per min.}$$

$$h = \frac{121.8 \times 144}{\text{density at 70 deg. fahr.}} = \frac{121.8 \times 144}{62.3} = 121.8 \times 2.30 = 282 \text{ ft.}$$

Head is the difference in level, measured in feet, between which flow takes place. If the velocity changes in passing through the system, the velocity head, $h = \frac{V^2}{2g}$, must be added to the suction and discharge heads to obtain the total head.

Example 54. — In Example 53 the diameter of the discharge pipe was 4 in., and the inlet pipe 6 in. Find the additional head caused by the change in velocity.

Solution. —

$$\text{Velocity head in discharge} = \frac{V_1^2}{2g}; \text{ velocity head in inlet} = \frac{V_2^2}{2g}$$

$$\text{Head caused by velocity change} = \frac{V_1^2 - V_2^2}{2g} = \frac{(6.72)^2 - (2.96)^2}{2 \times 32.2} = 0.565 \text{ ft.}$$

$$V_1 = \frac{\text{cu. ft. of water per sec.}}{\text{area of 4 in. discharge pipe}} = \frac{0.5950}{0.0884} = 6.72 \text{ ft. per sec.}$$

$$V_2 = \frac{\text{cu. ft. of water per sec.}}{\text{area of 6 in. inlet pipe}} = \frac{0.5950}{0.2006} = 2.96 \text{ ft. per sec.}$$

Suction is the pressure produced by the weight of the atmosphere, acting to force a liquid into a space wherein a partial vacuum exists because of the removal of the fluid that originally filled the space.

The **indicated water horsepower** is found from the data obtained from indicator diagrams taken from the water end of the pump and worked up in the manner explained in Art. 390, page 388. Care should be taken to use an indicator having a piston properly fitted for use with water, otherwise incorrect results will be obtained on account of leakage past the piston. The shape of the indicator diagrams taken from a steam driven pump is nearly rectangular for both the steam and water ends, and the inlet and discharge lines are wavy because of the imperfect action of the valves.

Thermal efficiency of a pump is generally expressed as the number of foot-pounds of work done by the pump per million B.t.u. consumed by the driving engine, and is called **duty**

Example 55. — The following data were taken during the test of a 30-million-gallon steam-turbine-driven reduction-gear centrifugal pump: pressures, barometer 29.22 in. mercury, steam 160.1 lb. per sq. in. gage; total head on pump, 140.94 ft. of water; quality of steam, 96.88 per cent; temperature of water pumped, 34 deg. fahr.; water pumped per twenty-four hours, 21.3 million gallons; vacuum on turbine 14.16 lb. per sq. in.; temperature corresponding to vacuum, 51.7 deg. fahr.; weight of condensate per hour, 9389 lb. Find the duty of the pump under conditions of test.

Solution. — According to definition

$$\text{Duty} = \frac{\text{Foot-pounds of work} \times 1{,}000{,}000 \text{ B.t.u.}}{\text{Weight of steam} \times \text{B.t.u. per pound of steam above exhaust pressure}}$$

$$= \frac{1{,}467{,}537{,}000 \times 1{,}000{,}000}{9389 \times 1149.7} = 136{,}200{,}000 \text{ ft. lb. per million B.t.u.}$$

$$\text{Ft-lb. of work per hour} = \frac{30{,}000{,}000 \times 8.33}{24} \times 140.94 = 1{,}467{,}537{,}000.$$

Weight of steam per hour = 9389 lb.

B.t.u. per pound of steam $= xL + h_1 - h_2 = 0.968 \times 854.3 + 342.5 - 19.78 = 1149.7$

L corresponding to 174.5 lb. per sq. in. abs. = 854.3 B.t.u.
h_1 corresponding to 174.5 lb. per sq. in. abs. = 342.5 B.t.u.
h_2 corresponding to 0.187 lb. per sq. in. abs. = 19.78 B.t.u.

505. Installation of Pumps. — The foundations of pumps must be rigid, with the suction pipe short, and as direct as possible. The suction lift should never be over 15 feet, and when pumping warm water the pump should be located below the supply, as otherwise the formation of vapor at low pressure destroys the vacuum and prevents the pump from lifting the water. The suction piping should be tight, to prevent destruction of the vacuum by air leakage. The piping for a centrifugal pump should be self supporting, and provision should be made for priming the pump when starting. The pump may be primed in the following ways:

1. Locating pump below water level of supply.

2. Using of a foot valve, Art. 506, with connection to discharge pipe, if the discharge pipe is left full of water.
3. Attaching a vacuum pipe or injector to the opening at top of casing, to remove air from casing, so that water will rise into pump through suction pipe.
4. Filling pipe casing with water from an external source. In this case, a foot valve must be used at foot of suction pipe.

A check valve should be placed on the discharge line of the centrifugal pump to prevent breakage from water hammer and a gate valve to control the capacity.

506. Foot Valve and Strainer. — The foot valve, Fig. 433, is used to hold the water in the suction or discharge pipe of a pump. When used in the suction pipe, it permits starting the pump without priming. It consists of a cast-iron body having a number of lift, or hinged, rubber-faced valves, opening against a spring. The upper end of the body is attached to the suction pipe, and when the pump is in operation the valves remain open; but when it is idle the weight of water in the pipe closes the valves and keeps the pipe full of water.

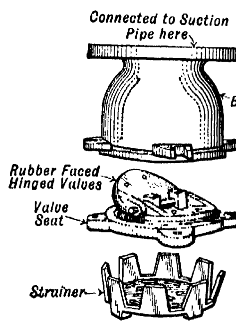


FIG. 433. — Goulds Foot Valve and Strainer.

The **strainer** consists of a frame supporting a removable wire-mesh strainer. It is attached to the suction pipe to keep out all foreign material which would clog the pump. When a foot valve is used, the strainer is attached to the lower part of the foot valve and is often an integral part of it.

507. Method of Testing a Pump. — The method of testing a steam pump is essentially the same as for a steam engine, with the addition of some suitable means of measuring the water delivered. For details egarding testing of pumps consult A. S. M. E. POWER TESTING CODE.

508. Valve Setting of Steam-driven Pumps. — To set the slide valve or valves of a steam-driven pump, the piston is usually placed midway between its extreme positions, and the valves placed centrally over both steam ports. When the lost motion allowed between the lug of the valve and the valve rod connection, which moves it, is made adjustable, the amount of the lost motion is made equal at each side of the lug by moving the nuts on the valve rod. The length of the stroke is controlled on most slide valve pumps by the amount of this lost motion The exact amount of lost motion to give the desired stroke must be found by trial.

509. Accumulator. — In the discharge pipe lines of pumps, where it is necessary to maintain nearly a constant pressure, an accumulator is used.

It consists essentially of a plunger of small area loaded with a large amount of iron. The plunger works in a cylinder connected directly into the pipe line. It rises and falls as water is forced in or passes out with slight change of pressure, storing water when the amount used is less than that supplied by the pump, and giving it out when the demand is greater than the supply. The pressure is thus maintained constant and the pump is allowed to run continuously. When the plunger rises to the extreme of its travel, it closes a valve on the pipe from the pump, and at the same time opens a by-pass between the suction and discharge pipes of the pump. While this valve is closed, water does not enter the plunger cylinder.

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Bulletin No. 2, Goulds Manufacturing Co.

REVIEW QUESTIONS AND PROBLEMS

1. Classify steam-driven piston pumps.
2. Describe the method of reversing the stroke on the Cameron pump.
3. How does a turbine centrifugal pump differ from a volute pump?
4. What is meant by a power-driven pump?
5. Explain the operation of a pump governor, and state why one is used.
6. Describe the construction and explain the operation of a single-tube injector.
7. Define: slip, water horsepower, pressure head, suction.
8. A 15.8-million-gallon vertical, triple-expansion plunger type pumping engine delivered 16.0 million gallons of water at 67 deg. fahr. through a total head of 298.7 ft. per twenty-four hours. Average steam pressure, 161.1 lb. gage; average quality of steam, 98.5; barometric pressure, 30.13 in. mercury; vacuum in exhaust pipe, 28.5 in. mercury; temperature corresponding to vacuum in exhaust pipe, 119 deg. fahr.; weight of steam condensed per hour, 9620 lb.; average i.hp., 871.8.

Calculate the duty in foot-pounds per million B.t.u.

9. Calculate the water horsepower and the mechanical efficiency of the pump in Problem 8.

CHAPTER XXV

CONDENSERS AND CONDENSER AUXILIARIES

510. Foreword. — *The primary object in operating engines and turbines "condensing" is to obtain a greater amount of useful work from a given weight of steam than could be obtained without the condenser.* The gain in power resulting from the use of a condenser is illustrated by Fig. 434 in which the

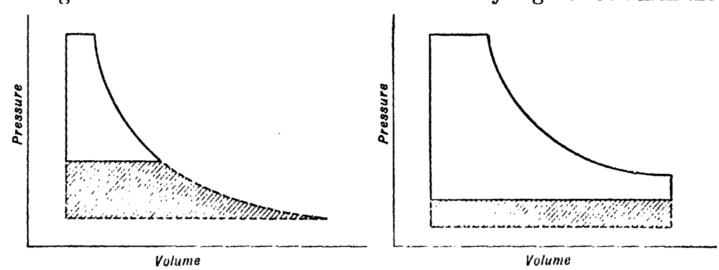


FIG. 434. — Diagrams showing Gain resulting from Condensing Operation.

full lines represent the theoretical diagrams for non-condensing operation. If the expansion could be continued or the back-pressure line lowered, as shown by the dotted lines, the area of the diagrams would be increased by the cross-hatched area. In steam engine practice the gain in power amounts to about 25 per cent for a vacuum of 26 inches of mercury, which is nearly the maximum for steam engine practice, because of the excessive size of cylinders required for the low pressure. The high cost, internal friction, and condensation losses of lower vacuums with reciprocating engines offset the gain in energy. An increase in power means an increase in economy, as shown by the curves in Fig. 435 for an engine operating condensing and non-condensing, with the

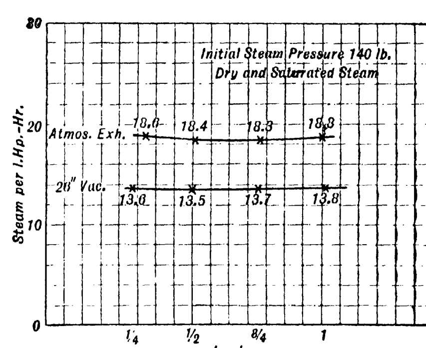


FIG. 435. — Water Rate Curves for Condensing and Non-condensing operation of a 21" x 22" Universal Unaflow Engine.

same initial steam pressure in each case. In marine practice the condenser is essential, to maintain the purity of the water in the boilers.

The advent of the steam turbine, operating either with high-pressure steam or with low-pressure steam from a reciprocating engine, has created a demand for a much higher vacuum than was required by the reciprocating steam engine. This has revolutionized condenser practice, and vacua corresponding to 28 or 29 inches of mercury are now common. The percentage of fuel saved for each of the last few inches in vacuum varies from 5 to 8 per cent, depending upon the type of turbine. In general, a point is reached between 28 and $28\frac{1}{2}$ inches vacuum at which the gain in economy is offset by the increased cost of condensing apparatus and cooling water.

511. Classification of Condensers. — Condensers may be roughly classified as (1) jet condensers, and (2) surface condensers. In the jet condenser the steam and water mingle, the steam being condensed by direct contact with water. In the surface condenser the steam and cooling water do not come in direct contact with each other, the heat of the steam being extracted by conduction.

512. Jet Condenser. — This class of condensers may be subdivided into:

(1) Low-level condensers $\begin{cases}\text{Standard}\\ \text{High-vacuum}\end{cases}$

(2) Barometric condensers

(3) Ejector condensers

Jet condensers are constructed on the parallel- or counter-current principle. In a **parallel-current condenser,** the steam, cooling water, and non-condensable vapors flow in the same direction, downward. Steam enters at the top or side of the condenser, and the condensing water immediately below. In a **counter-current condenser,** the steam enters the lower part of the condenser, and rises through the falling condensing water, which enters at the side near the top of the condenser. Air and other non-condensable vapors are withdrawn near the top of the counter-current condenser, after they have passed through the cooling water, and at the bottom of the parallel-current condenser, from a location directly over the surface of the water.

513. Standard Low-level Jet Condenser. — In this condenser, Fig. 436, the cooling water and a portion of the air are removed by some form of pump known as a wet-vacuum pump. The condensing water enters at the top of the pear-shaped condensing chamber and flows through a spray head which breaks the stream of water into a fine spray. Exhaust steam enters from the right and, mingling with the condensing water, gives up its latent heat of evaporation to the water. The condensed steam, water, and air are removed from the bottom by the wet-vacuum pump. The condensing water rises into the chamber, as a result of the vacuum maintained by the wet-vacuum pump, when the suction head is below 18 feet. In this type of

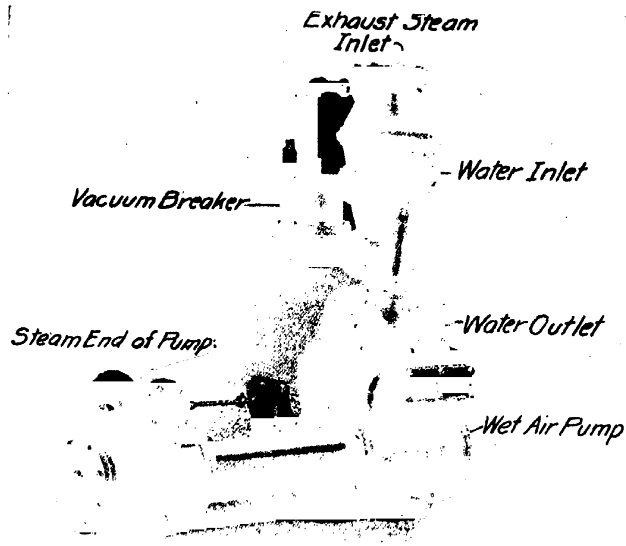


FIG. 436a. — Standard Low Level Jet Condenser — Outside View.

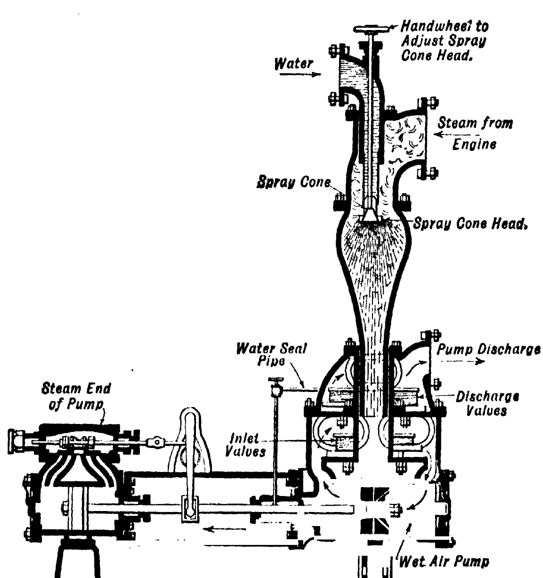


FIG. 436b. — Standard Low Level Jet Condenser — Sectional View.

condenser a vacuum-breaking device is used; it consists of a float located in a chamber attached near the top of the condensing chamber, and below the exhaust connection. The float opens a valve to the atmosphere, when the water enters faster than the pump removes it, and thus breaks the vacuum which prevents water from backing up into the exhaust pipe of the engine. This condenser has a low first cost, and is used with reciprocating steam engines where the vacuum required is not above 24 to 26 inches of mercury.

514. Barometric Condenser. — The barometric jet condenser, Fig. 437, differs from the standard jet condenser in that the condensing chamber is

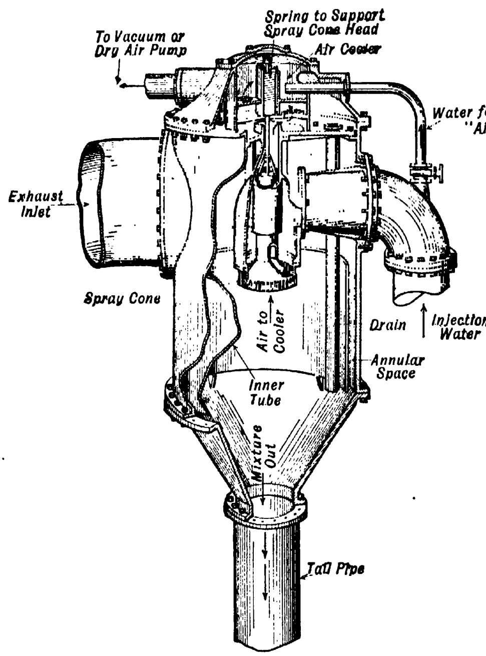


FIG. 437. — Alberger Barometric Jet Condenser — Internal View.

elevated about 35 feet above the discharge; the water is then discharged through the **tail pipe** without the aid of a pump. Water stands in the tail pipe to a height corresponding to the vacuum; this height would be 34 feet for a 30-inch vacuum.

The figure shows a section through the Alberger barometric condenser, with all equipment clearly named. Steam enters at the left and divides in two streams, one passing to the inner chamber and the other through the annular space surrounding the inner chamber. Cooling water is pumped through the **injection pipe** and passes through a **spray nozzle,** where it is

broken up into a fine spray. The cone forming the spray nozzle is hung from a spring, and automatically adjusts itself to the water supply. The water meets the steam at the lower end of the annular ring, condensing the steam and having its temperature raised above what it would be without the annular ring. The air not carried out by the cooling water collects

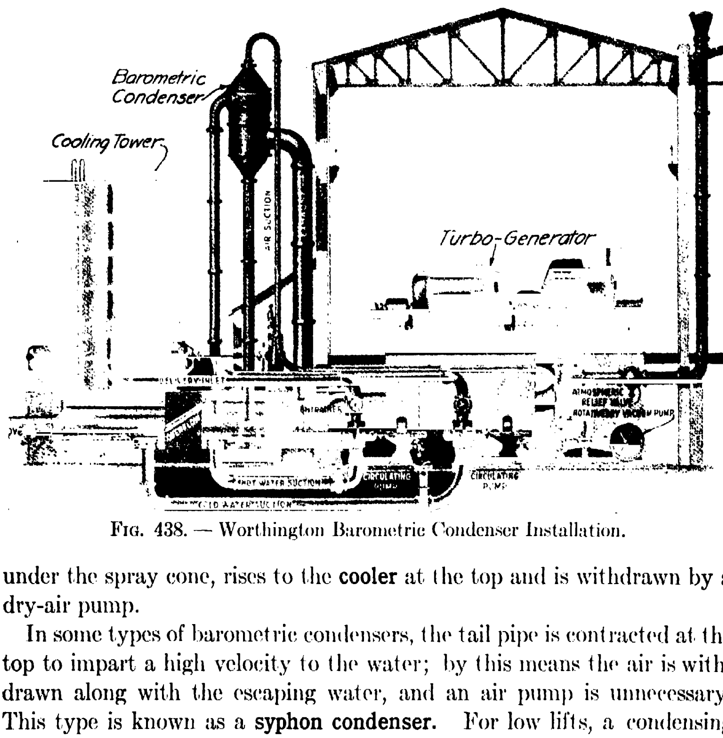


FIG. 438. — Worthington Barometric Condenser Installation.

under the spray cone, rises to the **cooler** at the top and is withdrawn by a dry-air pump.

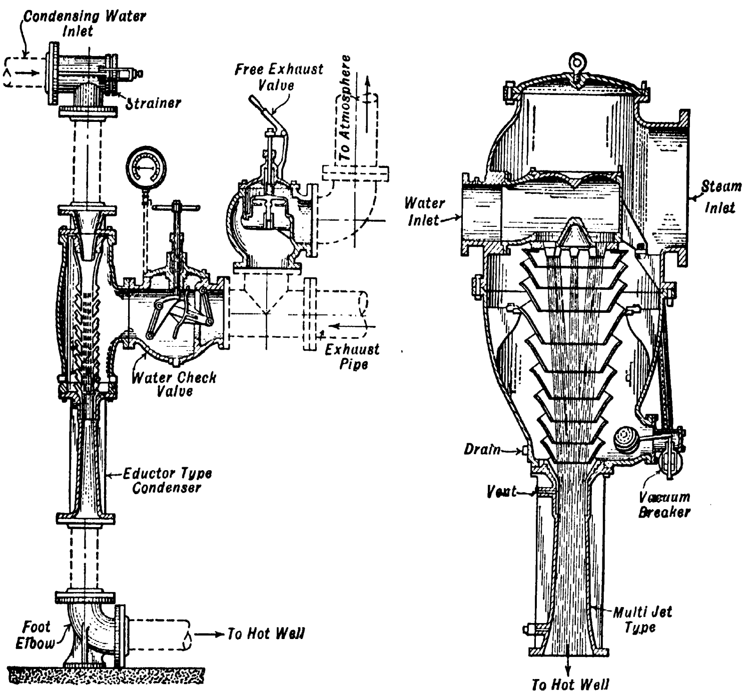
In some types of barometric condensers, the tail pipe is contracted at the top to impart a high velocity to the water; by this means the air is withdrawn along with the escaping water, and an air pump is unnecessary. This type is known as a **syphon condenser.** For low lifts, a condensing water pump is not necessary, as the injection water can be raised to a height corresponding to the vacuum.

The barometric condenser is capable of maintaining a vacuum of 27 inches, but requires a large amount of water which must be handled by the condensing water pump. A typical installation is shown in Fig 438.

515. Ejector Condenser. — The principle of operation of this condenser is that the momentum of flowing water ejects the discharge without the aid of a pump. Exhaust steam enters the ejector, Fig. 439, at the right and surrounds a central tube, in which are located numerous passageways. The cooling water enters the nozzle at the top and, because of the shape of the nozzle, attains considerable velocity. In flowing past the openings this water produces a vacuum, which causes the steam to flow through the

openings and mix with the condensing water. The cooling water enters continuously at the top, because of the vacuum, and sufficient velocity is given the jet to discharge the combined mass of condensed steam, cooling water and air, against the pressure of the atmosphere.

There is about 3 feet of pipe above the nozzle and at least 2 feet below the discharge, which should be water-sealed. The vacua attained are



(*a*) *Standard Type* (*b*) *Multi-jet Type*

FIG. 439. — Ejector Condensers.

from 20 to 25 inches for the single jet types, and up to the highest vacua for the multi-jet type.

516. High-vacuum Jet Condenser. — This condenser is of the low-level type. The jet condensers previously described are not suitable for producing the high vacua necessary for turbine operation, because the air pumps used are not adapted to handle large quantities of air. Several representative types of high-vacuum condensers are shown in Figs. 440 and 442. The former is a **rectangular rain-type** condenser, in which the injection water is introduced, at the top near one end, into an extended trough or pan, from which it overflows through numerous short tubes,

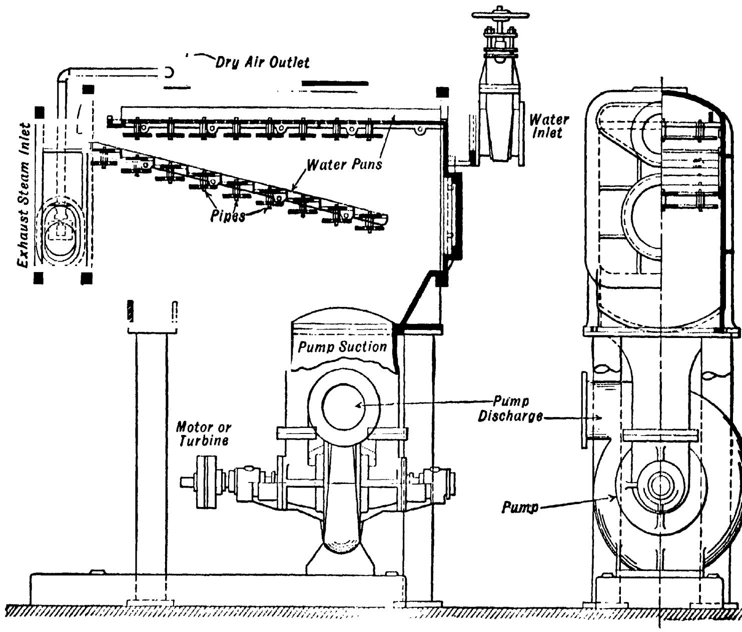


FIG. 440. — Wheeler Rectangular Rain-type Jet Condenser.

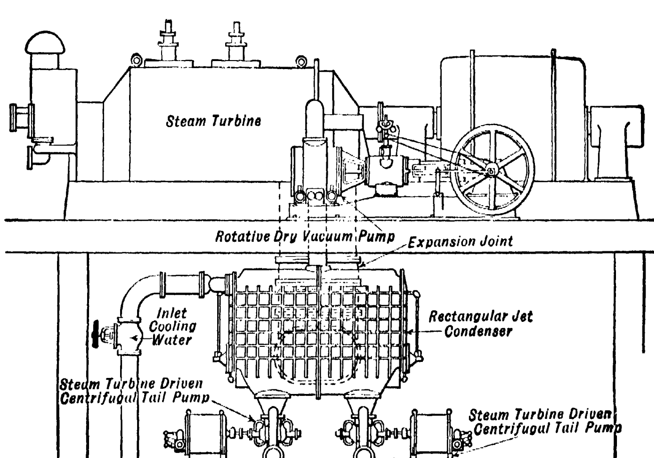


FIG. 441. — Wheeler Rectangular Jet Condenser and Installation.

falling into a second pan provided with similar overflow pipes, and finally into the lower part of the shell and thence to the vacuum pump.

Exhaust steam enters at the left, passes into the shower of water, is condensed and passes out with the condensing water. The air and mixed vapor pass to the top and are withdrawn by an air pump of suitable con-

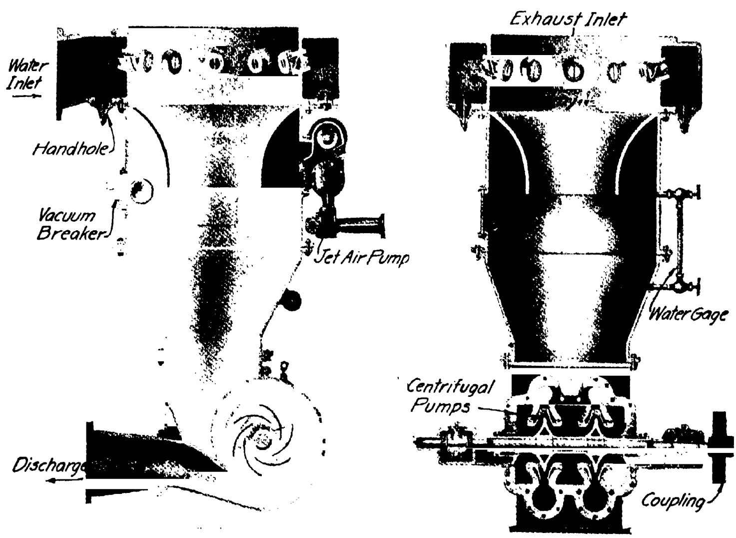


FIG. 442. — Elliott Erhart Low Level Jet Condenser. — Sectional Views.

struction. A vacuum-breaker is located at the left of the chamber. A centrifugal pump removes the water from the condenser, which is capable of producing a high vacuum on account of the thorough mixing of steam and water and the use of a separate air pump. A typical installation is shown in Fig. 441.

The **Elliott-Erhartt low-level jet condenser,** shown in Fig. 442, is a type much used for high-vacuum work because of its low first cost. Condensing water enters at the left side near the top and is distributed around the circumference by a chamber, from which **spiral nozzles** discharge the water into the **condensing cone.** Steam enters from the top and is thoroughly mixed with the condensing water from the nozzles. The cooling water enters by virtue of the vacuum maintained within the condenser, provided the suction head is less than 18 feet. The mixture of condensed steam and cooling water is removed from the bottom of the condenser by a centrifugal pump, generally driven by a steam turbine. The air and vapor are removed by an **ejector air pump** shown on the right. A vacuum breaker is shown at the left, just below the condensing cone. This condenser is especially

useful in places where the condensing water is not suitable for surface condensers. Its cost of operation may, however, be high, because of the power required to remove the large amount of condensing water.

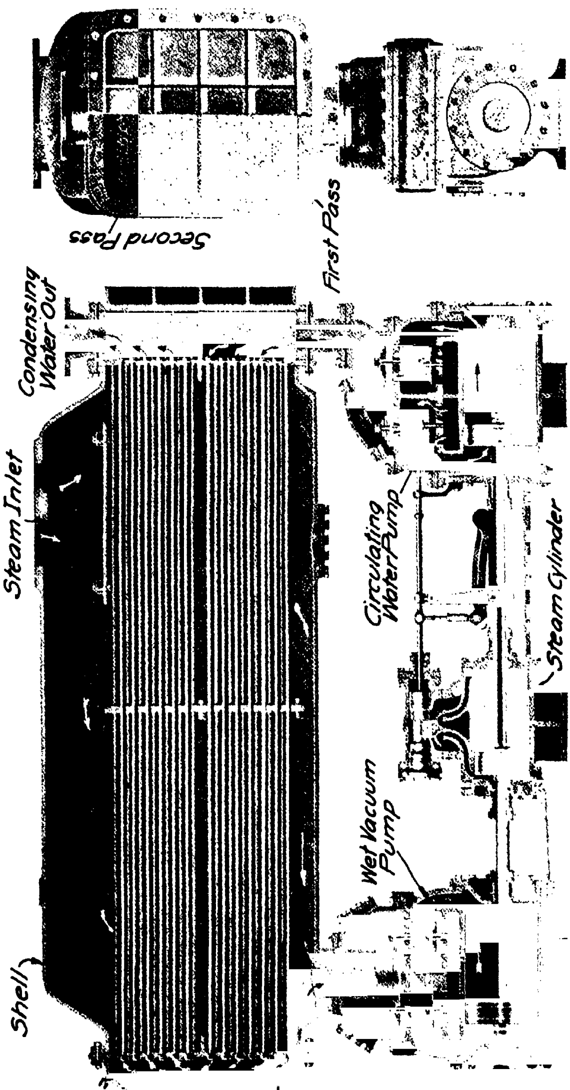


FIG. 443. — Wheeler Admiralty Surface Condenser Mounted over Combined Air and Circulating Pumps.

517. Surface Condensers. — There are two general types of surface condensers, namely, (1) those in which the cooling water is inside the tubes

and the exhaust steam on the outside, and (2) those in which the steam is inside and the water outside the tubes, known as the water-works type. The first type is most common and generally operates on the **counter-current principle**; that is, the water and steam flow in opposite directions.

518. Wheeler Standard Surface Condenser. — This condenser, Fig. 443, which is adapted for steam engine service, consists of a cast-iron shell closed at each end by a head. Between each head and the shell is located a **tube sheet,** into which the tubes are fastened, as shown in the enlarged view. Packing is placed in the tube sheet surrounding each tube, and the **ferrule** is tightened to prevent leakage. The tubes are made of copper or brass, and are supported by a plate at the middle of their length. Water enters

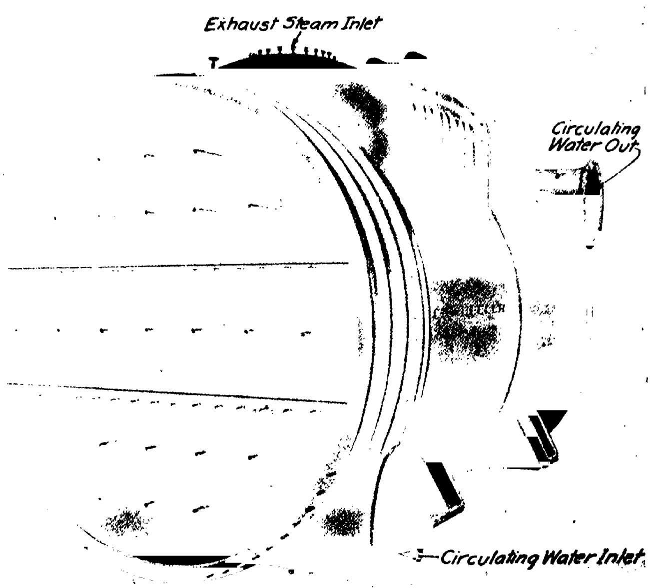


Fig. 444. — Surface Condenser for 10,000-kw. Turbine — Three-pass Design.

at the bottom right, passes to the left in the lower bank of tubes, and returns to the top right in the upper bank of tubes, thus passing the length of the condenser twice. Steam enters at the top and strikes against a **distributing baffle** which prevents excessive short-circuiting of the steam. The condensed steam and air are withdrawn at the lower left-hand side by a wet-air pump.

519. High-vacuum Surface Condenser. — The surface condenser previously described is suitable for a vacuum of about 26 inches. To obtain a vacuum of about 28 or 29 inches the following conditions must be fulfilled: (1) the air entering the condenser must be rapidly removed, because when

allowed to accumulate it acts as a blanket about the tubes and interferes with the heat transmission; (2) the steam distribution must be made effective, by omitting tubes, Fig. 444, thus forming **steam lanes,** which conduct the steam to short banks of tubes and prevent the formation of dead air spots; (3) the air must be removed by a separate air pump, with an air connection located at a point where the air is the coolest and has the greatest density and hence the smallest volume; (4) baffles must be inserted between the banks of tubes to drain off the condensate and prevent flooding of the tubes; (5) condensing water should pass through the condenser with the least possible friction, but at a velocity consistent with high efficiency.

Condensers of this type are made in sizes containing 70,000 square feet of condensing surface.

520. Conditions Necessary for Heat Transfer. — The transfer of heat from steam to the cooling water in a surface condenser depends (1) upon the condition of the tube surface, (2) upon the proportion of air in the steam and the water, and (3) to some extent upon the material of which the tubes are made. For ordinary types of surface condensers, from 250 to 300 B.t.u. per hour are transmitted per square foot of tube surface for each deg. fahr. rise in temperature of the circulating water. In the high vacuum condensers, this value may amount to 600 or 800 B.t.u. per square foot per deg. fahr. per hour. Each square foot of cooling surface will condense about 10 pounds of steam per hour, for ordinary condensers at 24 to 26 inches vacuum. For small turbine installations, $2\frac{1}{2}$ to 4 square feet of cooling surface per kilowatt are provided, and for large plants with high efficiency condensers, 1 to $2\frac{1}{2}$ square feet per kilowatt rated capacity.

521. Comparison of Types. — Jet condensers have low first cost, occupy small space, frequently require more pump capacity, and have more air to remove, thus requiring larger air pumps. Surface condensers occupy large space, give high efficiency, and provide pure feedwater where the condensing water is impure.

522. Elementary Theory of Condensers. — The vacuum within a condenser is customarily referred to a 30-inch barometer. The height of the standard barometer, at sea level and latitude 45 degrees is 29.92 inches when the temperature of the mercury is 32 deg. fahr. For a 30-inch barometer the temperature of the mercury is increased to 58.4* deg. fahr. To correct a mercury column for change in temperature, the following equation may be used,

$$h = h_1 [1 - 0.000101 (t_1 - t)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (105)$$

in which h = height of mercury column corrected to temperature t.
h_1 = observed height of mercury column, inches.
t_1 = observed temperature of mercury column, deg. fahr.
t = temperature to which column is to be referred, deg. fahr.

* This value is sometimes taken as 58.1 deg. fahr.

Example 56. — The height of mercury in a manometer used to measure vacuum is 28.52 in., at a temperature of 80 deg. fahr., barometer reading 29.85 in. at a temperature of 42 deg. fahr. Determine the vacuum, referred to a 30-in. barometer.

Solution. — Vacuum corrected, $h = 28.52\,[1 - 0.000101\,(80 - 58.4)]$
$= 28.46$ in. mercury.
Barometer corrected, $h = 29.85\,[1 - 0.000101\,(42 - 58.4)]$
$= 29.89$ in. mercury.

Absolute pressure in inches mercury $= 29.89 - 28.46 = 1.44$ in. mercury.

Vacuum referred to a 30-in. barometer $= 30.00 - 1.44 = 28.56$.

523. Condenser Pressure. — *The pressure existing within a condenser is made up of the pressure of the air plus that of the water vapor.* This is in accordance with **Dalton's law**, which states that in *a mixture of gas and vapor enclosed in a given space, the total pressure is equal to the sum of the partial pressures which each gas would exert if occupying the space alone.* As the vacuum increases, the proportion by weight of air to vapor increases for a given air pressure, and there is a corresponding increase in the amount of air that must be removed to maintain the vacuum.

Example 57. — The absolute pressure in a condenser is 2.03 in. of mercury, and the temperature of the air-vapor mixture is 100.6 deg. fahr. Find the pressure produced by the air within the condenser.

Solution. — From the Steam Table, the pressure of the vapor at 100.6 deg. fahr. is 1.97 in. mercury, and therefore, by Dalton's Law,

$$P_c = P_a + P_v, \quad \text{or} \quad P_a = P_c - P_v$$

in which P_c = condenser pressure, in. mercury = 2.03
P_a = air pressure, in. mercury.
P_v = vapor pressure = 1.97 in. mercury.
$P_a = P_c - P_v = 2.03 - 1.97 = 0.06$ in. mercury.

524. Weight of Cooling Water Required by a Condenser. — The quantity of cooling water per pound of steam condensed depends upon the degree of vacuum obtained and the inlet and outlet temperatures of the cooling water. The weight of cooling water per pound of steam varies from 25 to 30 pounds for vacua of 25 to 26 inches of mercury, and increases to above 50 pounds for high vacua.

Neglecting the effect of radiation and leakage, the heat absorbed by the cooling water will equal that given up by the steam in the condenser. *For a surface condenser,* the above relation may be written.

$$W = \frac{xL + h - h_c}{h_2 - h_1} = \frac{xL + h - (t_c - 32)^*}{t_2 - t_1} \quad . . . \quad (106)$$

* The second form of this equation holds for condensers, provided the influence of the entrained air on the heat content of the exhaust steam is neglected, and the mean specific heat of the water at condenser temperatures is taken as unity. W is usually increased from 5 to 15 per cent, to allow for cooling of the air-vapor mixture and inefficient absorption of heat.

in which W = weight of cooling water per hour per pound of steam condensed, lb.

x = quality of exhaust steam; assumed as 90 per cent or figured from Equation (92), page 406.

L = latent heat at the absolute pressure in the condenser, B.t.u.

h = heat of liquid at the absolute pressure in the condenser, B.t.u.

h_c = heat of liquid at the temperature of the condensate (t_c), B.t.u.

t_c = from 5 to 20 deg. fahr. below the temperature of the exhaust, deg. fahr.

h_2 = heat of liquid at the temperature (t_2) of the outlet condensing water, B.t.u.

t_2 = from 10 to 25 deg. fahr. below the temperature corresponding to the absolute pressure in the condenser. For an average value 15 deg. fahr. may be used.

h_1 = heat of liquid at inlet temperature (t_1).

t_1 = temperature of inlet condensing water.

Example 58. — The following data were taken from the test of a condenser attached to a 30,000-kw. turbine: Duration of test, 3 hr.; steam condensed per hour, 317,000 lb.; steam pressure, 237 lb. sq. in. abs.; superheat, 155 deg. fahr.; vacuum, 28.17 in. mercury; quality at exhaust, 0.90; initial temperature of cooling water, 76.5 deg. fahr. Find weight of cooling water per hour. Temperature of condensate, 91.3 deg. fahr.; barometer, 29.93 in. mercury.

Solution. — Using the second form of Equation (106) with proper substitutions,

$$W = \frac{xL + h - (t_c - 32)}{t_2 - t_1} = \frac{0.90 \times 1038.8 + 64.93 - (91.3 - 32)}{86.9 - 76.5}$$

$$= \frac{940.55}{10.4} = 90.5 \text{ lb. water per pound of steam.}$$

L at 1.76 in. mercury, abs. = 1038.8 B.t.u.; h at 1.76 in. mercury, abs. = 64.93 B.t.u.; t_c = 91.3 deg. fahr.; t_2 = 96.9 − 10 = 86.9 deg. fahr.; t_1 = 76.5 deg. fahr.

Total weight of cooling water per hour = 90.5 × 317,000 = 28,688,500 lb.

For a jet condenser, the final temperature of the cooling water equals that of the condensate, and t_c varies from 5 to 20 deg. fahr. below the temperature corresponding to the vacuum, because of the pressure exerted by air and similar gases. Making these changes in Equation (106), there results:

$$W = \frac{xL + h - h_c}{h_2 - h_1} = \frac{xL + h - (t_c - 32)}{t_c - t_1} \quad . \quad . \quad . \quad . \quad (107)$$

Example 59. — Using the data of Example 58, find the weight of cooling water required if a jet condenser were used.

Solution. — Using equation (107), second form, and making proper substitutions,

$$W = \frac{xL + h - (t_c - 32)}{t_c - t_1} = \frac{0.90 \times 1038.8 + 64.93 - (91.3 - 32)}{91.3 - 76.5}$$

$$= \frac{940.55}{14.8} = 63.5 \text{ lb. water per pound of steam.}$$

Total weight of cooling water per hour = 63.5 × 317,000 = 20,129,500 lb.

525. Condenser Auxiliaries and Accessories. — The principal auxiliaries used with condensers are water-circulating pumps, dry- and wet-air or vacuum pumps, relief valves, expansion joints, gate valves, and water-cooling apparatus.

526. Circulating Pumps. — The types of circulating pumps used with condensers are direct-acting and centrifugal. The latter are now being used almost entirely because of their simplicity.

527. Air Pumps. — The function of the air pump is to maintain the vacuum formed by the condenser. Non-condensable gases enter with the

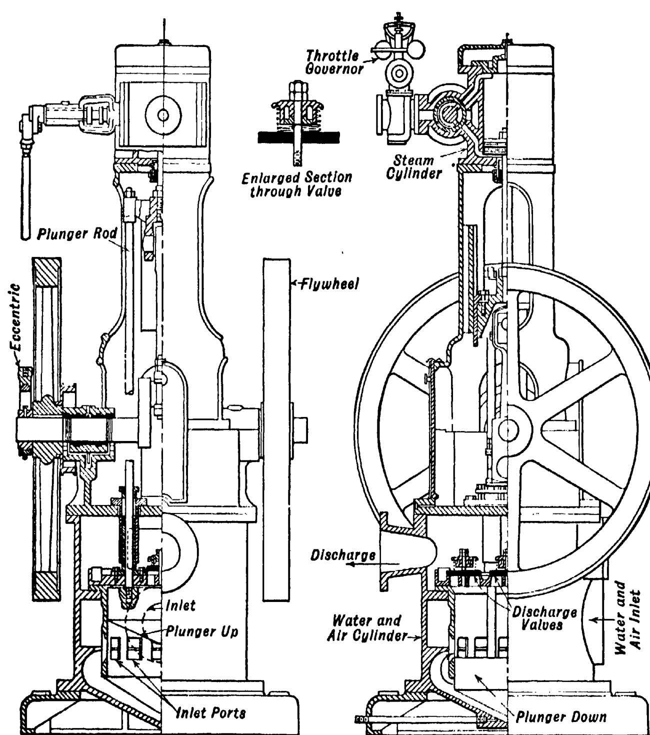


FIG. 445. — Edwards Air Pump.

steam and leak in through the various parts; in the case of a jet condenser they also enter in solution with the cooling water. An air pump should be capable of removing a large volume of air under all conditions of operation, for highest condenser efficiency. Four different types of air pumps are commonly used, namely, reciprocating, rotary displacement, centrifugal entrainment, and steam-ejector vacuum pumps.

528. Wheeler-Edwards Air Pump. — This pump, which is shown in section in Fig. 445, is a wet-air pump consisting of a plunger driven by a steam piston located directly above the pump. The pump has no inlet valves, as the inlet is through ports uncovered by the piston when on its down stroke. Water of condensation flows continuously, by gravity, from the condenser into the base of the pump. Upon descent of the conical plunger, this water is projected at a high velocity through the ports into the working barrel. As soon as the ports are uncovered by the piston, air is also drawn into the barrel, along with the water, because of the vacuum formed by the descending bucket. The plunger, upon rising, discharges the mixture through the valves at the top of the barrel, the air passing out first.

The valves are made of rubber, the barrel of brass, the valve plate of composition, and the pump rods of Tobin bronze.

This pump is capable of producing a 28-inch vacuum when used in connection with a **dry-tube surface condenser.** It is especially adaptable to marine work.

529. Mullan Displacement Vacuum Pump. — The Mullan pump is double-acting, operating upon the wet-vacuum principle and without suction valves. The piston, at the end of its suction stroke, uncovers a series of ports distributed around the middle of the cylinder, through which the vapors are drawn into the cylinder. The discharge valves are of the **Gutermuth type,** Fig. 446, consisting of a strip of phosphor bronze coiled at one end and attached to the valve stem. They are mounted on a plate, or **deck,** attached to each end of the pump barrel. The condensate enters the cylinder at the top or side, and is forced through the discharge valve, together with the air, on the discharge stroke of the piston.

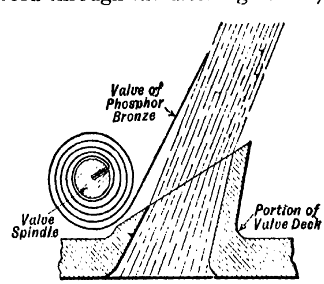


FIG. 446. — Gutermuth Flexible Metallic Valve.

The water cylinder, Fig. 447, is lined with hard brass. The piston is of iron, brass covered, cast very light and with ample surface to ensure minimum pressure on the liner. The actual bearing surface of the piston on the liner consists of a number of end-grain **lignum vitae plugs,** which are turned off slightly larger in diameter than the piston cover. This combination of lignum vitae and brass, with water lubrication, is virtually wear proof and has an important bearing on the durability of the Mullan pump. A series of **water grooves** turned in the piston surface make it vacuum tight without the use of piston rings. The piston rod of Tobin bronze is secured in the piston by a taper end, lock nut, and pin. It works through a long double stuffing box containing a water seal in addition to the packing, eliminating all possibility of air leakage at that point.

The steam end is of the most modern design. The valve gear comprises an eccentric, working a "D" slide valve for the smaller pumps, or a balanced piston valve with self-adjusting snap rings for the larger pumps.

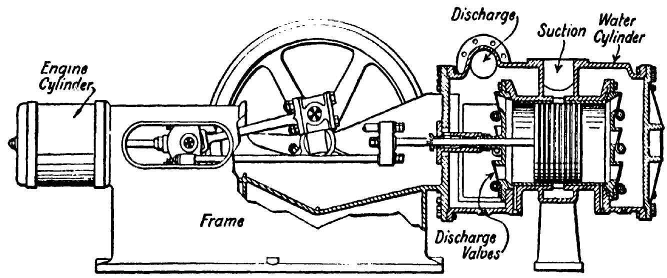


FIG. 447. — Mullan Air Pump.

A throttling governor with a safety stop ensures against overspeeding from any cause.

This pump is suitable to maintain a vacuum within a small fraction of an inch of the theoretically perfect vacuum.

530. Rotative Dry-vacuum Pump. — This type of vacuum pump resembles an air compressor in mechanical construction and operation. It is capable of producing a high vacuum, operates with high efficiency, and is used where a high degree of vacuum is essential as with condensers attached to turbines.

The Wheeler steam-driven pump shown in Fig. 448 has the air and steam cylinders arranged in tandem with the steam cylinder nearer the crank. The

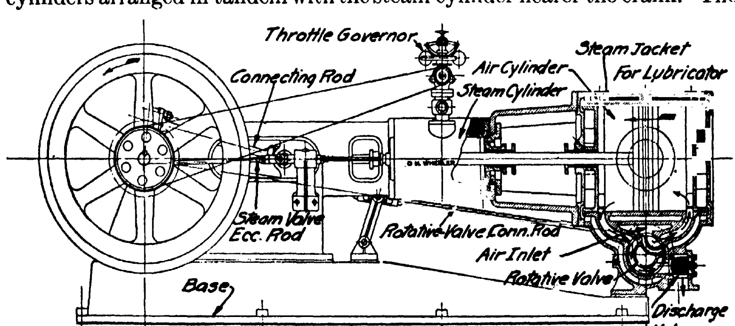


FIG. 448. — Wheeler Rotative Steam-driven Dry Vacuum Pump.

steam cylinder has a piston valve operated by an eccentric connected to the main shaft. The air cylinder is completely water-jacketed and the semi-rotative inlet valve is mechanically driven from the crank shaft, by means of an eccentric operating a crank attached to the valve rod; while the discharge valves are of the poppet spring-loaded type.

For the position of the air piston and valve shown, the piston is moving to the left compressing the gases and discharging them, at a pressure slightly above that of the atmosphere, through the poppet discharge valves. The movement of the piston to the left lowers the pressure on the right of the piston and the gases from the condenser flow in to fill the space behind the piston. When the piston reaches the end of the stroke, the rotative valve is moved to close the suction port and open an auxiliary passage connecting each end of the cylinder. This equalizes the pressure and, by eliminating the effect of the clearance space, prevents the loss in efficiency which would result from the re-expansion of the gases in the clearance space. The cycle of events for a typical rotative dry-vacuum pump is shown in Fig. 449.

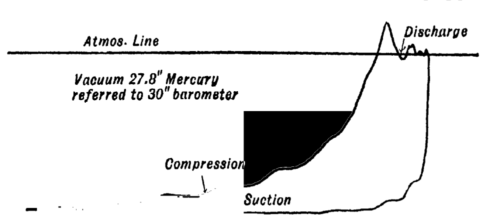


FIG. 449. — Diagram from Dry-vacuum Pump.

Valves of the multi-disk type are used on many rotative dry-vacuum pumps and are claimed to operate at high efficiency.

531. Centrifugal Entrainment Pump. — One type of this pump, used on the Westinghouse-Leblanc low-level jet condenser is shown in Fig. 450, in

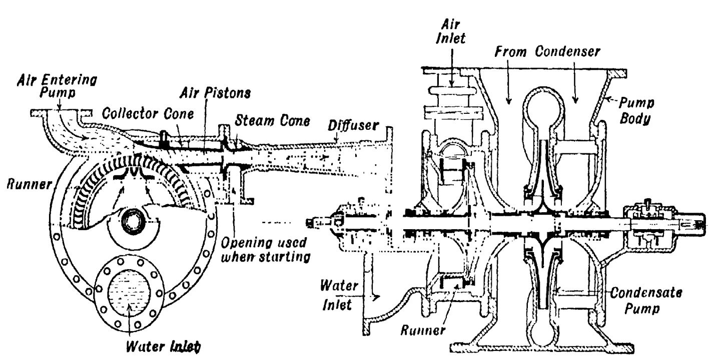


FIG. 450. — Leblanc Centrifugal Entrainment Vacuum Pump.

which the rotors of the air and condensate pumps are mounted on the same shaft. The air pump consists of a stationary nozzle, rotating vanes located in a casing, and an inlet and discharge pipe. The operation is as follows: water for operating the pump enters the inner chamber, surrounds the shaft, and flows out through the nozzle. The rotating vanes on the impeller rotate clockwise and cut off layers of water, which are projected continuously into

the cone, at high velocity. Air is caught, or **entrained,** between successive layers of water and is forced at approximately atmospheric pressure into the discharge cone. The high velocity of the water pistons is converted into pressure by means of the diffusing cone, which permits discharging against pressure. A steam ejector is used to put the condenser into service quickly.

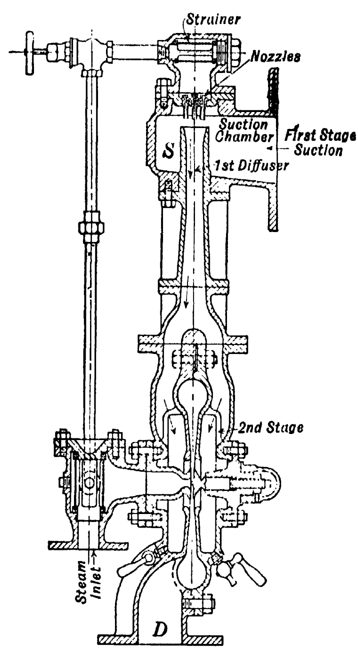


FIG. 451. — Cross-sectional View of Radojet Vacuum Pump.

532. Ejector Vacuum Pump. — This is the most recent type of high vacuum pump. It represents the highest development in vacuum producing apparatus and is being used by nearly all manufacturers of high-grade equipment.

The Radojet pump shown in Fig. 451 is a typical pump. It operates on the dry-air principle, and is a substitute for air pumps of the reciprocating, rotative, or hydraulic entrainment type. It has no moving parts; its weight and its space requirements are small; its efficiency is high and does not change after long periods of operation; it does not require lubrication or attention during operation; and it is simple and rugged in construction.

The Radojet consists of two steam ejectors working in series, the upper ejector being called the first stage, and the lower, the second stage.

Live steam is delivered to the steam inlet and, in passing through the strainer, divides, a part going to the first stage and the remainder to the second stage of the pump.

The part going to the first stage rises through the small pipe at the left to the auxiliary steam valve and first-stage strainer. It then passes through the first-stage expansion nozzles across the upper suction chamber of the first-stage ejector, which is connected with the condenser through the suction opening. The steam expands in the nozzles, leaving with a high velocity, and in passing across the suction chamber entrains the air and vapors to be compressed and removed.

The mixture of air and steam passes into the **upper diffuser,** from which it is discharged, at a higher absolute pressure than that of the air entering at S, into a double passage communicating with the suction chambers of the

second stage. These two suction chambers are annular, giving the commingled fluid a large entrainment surface.

At the same time, steam is delivered from the strainer into the passage, leading to the second stage, which communicates with the **annular expansion nozzle** formed between two circular disks. One disk is adjustable to vary the cross section of the second-stage nozzle passage, thereby changing the expansion ratio of the steam.

The steam delivered radially by the annular nozzle expands, leaving the nozzle as a jet of high velocity in the form of a sheet. This steam, in passing across the second-stage suction chambers, takes the water and air mixture coming from the first stage and carries it into the second-stage annular diffuser, thereby compressing the mixture to atmospheric pressure and discharging it into the casing, which has a discharge opening at D.

The steam nozzles and diffusers are designed to give the highest overall efficiency. The steam nozzles of the first stage are bronze, and those of the second stage bronze and special steel. The diffusers are bronze, and in the smaller sizes form part of the casing, while in the larger sizes they are secured to a cast-iron casing, by bolts and a ground joint.

The steam consumption of this pump is low, and the thermal efficiency close to 95 per cent.

A typical installation of an ejector pump is shown in Fig. 442, page 530.

533. Exhaust Connections. — The exhaust connections to a condenser should contain (1) an atmospheric exhaust relief valve, to give free exhaust to the atmosphere should the vacuum fail; (2) an expansion joint, to make a flexible link between the prime mover and condenser; and (3) a gate valve to isolate the prime mover when necessary.

Atmospheric Relief Valve. — A typical valve of this type is shown in Fig. 452. The relief valve is placed in a branch taken from the main exhaust line, leading to the atmosphere. As long as the vacuum is maintained in the condenser, the relief valve remains tightly closed and is water-sealed to prevent leakage. As soon as the vacuum is lost, the valve promptly opens and the engine exhausts to the atmosphere until the vacuum is restored, when the valve closes, automatically. A dash pot is provided to permit the valve to close quietly, and a lip is placed around the valve seat for water-sealing.

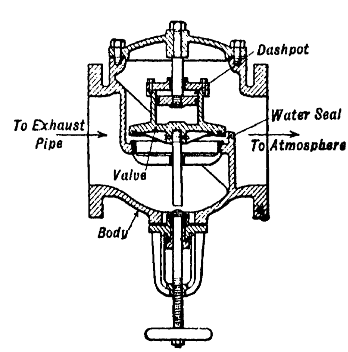


Fig. 452. — Atmospheric Relief Valve.

Expansion Joint. — This joint is generally made of corrugated copper of

suitable thickness, with the ends flanged over and fitted into cast-iron flanged rings, accurately faced and drilled.

Owing to the form of the corrugations, it can be made very short, thereby minimizing headroom. The expansion joint is often omitted on large condensers and the expansion allowed for by supporting the condenser on heavy springs.

Turbine Exhaust Valve.—A gate valve is sometimes required between the turbine and condenser. By combining the functions of such a gate valve, and atmospheric outlet, it is possible to provide an opening for the atmospheric relief valve, and at the same time minimize the headroom required; this device may also obviate the necessity of excavating several feet in a basement, in order to place the condenser under the turbine. Such a valve, made by the C. H. Wheeler Mfg. Co., has a single gate with the face and seat made of babbitt metal, accurately machined. When open, the gate is supported on rollers running on guides which lift it off its seat, thereby making the gate practically frictionless, except at the point of seating, where the guides slope downward to permit it to reach its seat. Wedge pieces are further provided, to maintain a positive seat.

The atmospheric relief outlet communicates with the inlet side of the gate, and, when the gate is closed, carries the exhaust steam to the atmospheric relief valve previously described.

534. Cooling Ponds and Towers. — Where a power plant or manufacturing plant is not located adjacent to, or within a reasonable distance of, a river, lake, or other source of natural water supply, the cold water required for condensers and for many other industrial purposes must be obtained by means of a **water re-cooling system.** It also frequently occurs that such natural water as is available is not suitable for the required purpose, because of the presence of free acid or sewage contamination, resulting in the rapid deterioration of the metal parts of the condensers or other apparatus through which the water flows. Also, the available water may contain considerable quantities of foreign matter which render it unsuitable for condensing or cooling purposes. In either one of these cases, some water re-cooling system must be employed. *All such systems depend upon the exposure of the warmed water to the evaporation and consequent cooling effect of the atmosphere.*

The process of re-cooling water is generally accomplished by either cooling ponds or cooling towers.

Of the two methods, the **cooling pond** is much the cheaper, provided an adjacent and suitable ground or roof area of sufficient extent, is available. The cooling results obtained from a cooling pond, however, are not equal to those obtained from a tower, either of the forced or natural draft type.

The simplest method of cooling water is to discharge it through a single pipe line to a pond of sufficient area, so that the water will be cooled to the

temperature desired, by contact with the atmosphere on the surface of the water. Such a pond, however, must be of large dimensions, and, to reduce the area required, it is necessary to add some device by which additional contact of the water with the atmosphere is obtained. This device consists of a spraying system, of which there are a number of types in use at the present time.

The most desirable features of any spraying system, as given by the C. H. Wheeler Mfg. Co., are as follows: (1) low initial cost, (2) low operating cost, (3) extensive cooling range, (4) low maintenance charge, (5) first-class materials and mechanical construction, (6) adjustability of the mechanism to produce either a fine or a coarse spray, as required by the weather conditions, (7) the use of a small pond area and of as small an amount of piping in the pond as possible, (8) the elimination of driftage, and (9) means by which the spraying mechanism may be quickly cleaned without interfering with the operation of the cooling system.

535. Spray Cooling System. — A cooling pond using sprays is shown in operation and, when not in operation, in Fig. 453. The latter shows the distributing pipe system and the arrangement of the spraying apparatus

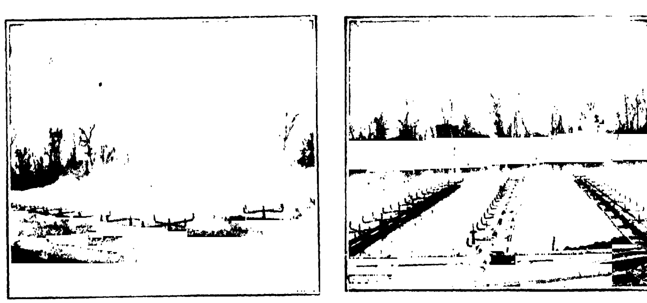


FIG. 453. — Spray Cooling Pond — In Operation and Idle.

over the pond. This type of cooling pond may be constructed of concrete, or it may be of natural soil having sod banks. Such ponds need not be deep, except when depth is required for storage purposes, a depth of 3 feet or even less being ordinarily sufficient.

The spray is produced by some type of nozzle, which aims to break the water up into a spray suitable for the weather conditions under which it is working. In all types, the shape of the nozzle is such that it gives to the water a whirling motion, thus producing more effective atomizing of the water.

Spray nozzles are constructed with (*a*) rigid heads, and (*b*) adjustable heads. A few typical nozzles of each class are shown in Fig. 454. The construction of each type is evident from the illustrations.

An adjustable spray head, which has given satisfactory service, was designed by Professor Carl C. Thomas and described in Trans. Am. Soc. M. E., 1917, page 625. This head consists of a cast-iron supporting base containing the water-entry opening and carrying a bronze tube wound spirally. This tube is held between the base and a cap, which fits the top, by a central bronze stem passing down through a close-fitting bushing in the base. This stem is movable and is operated by a bell crank, having an extended vertical arm which gives accurate control of the position of the

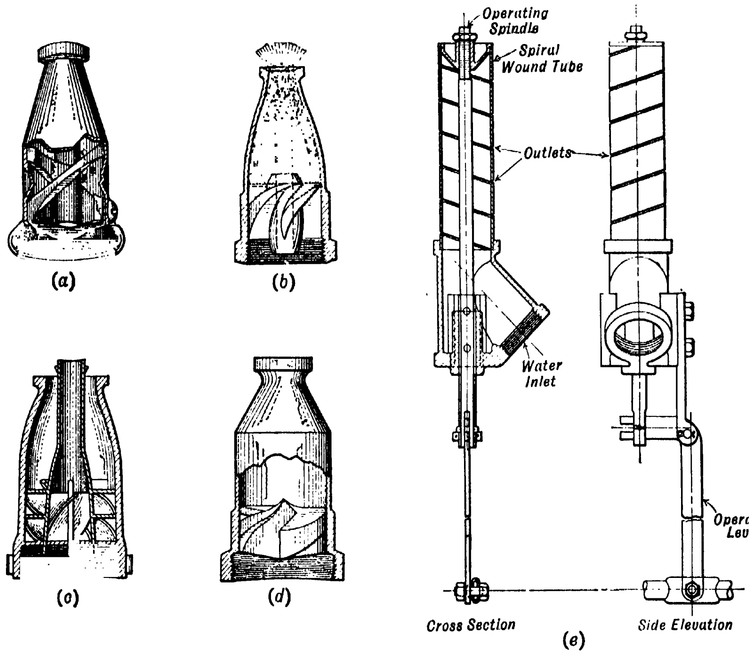


Fig. 454. — Rigid Head and Adjustable Head Spray Nozzles.

stem. By moving the stem slightly, the opening of the spiral, through which the water is discharged, is varied. The result of the motion, which is made by a system of levers located at some convenient point on the bank of the pond, is that the fineness of the film of water is varied. Water is discharged from the head in a continuous stream, during the time of operation, with an upward inclination. The water film spreads, and becomes thinner on account of its increasing diameter. Finally, a point is reached where the *surface tension* of the water is overcome, and the sheet of water breaks into a fine spray, a mist, or an infinite number of small drops, depending upon the adjustment of the size of the spiral opening.

The nozzles and spray heads are attached to a system of piping, into

which the cooling water is pumped at a pressure varying from 3 to 8 pounds, depending upon the type of nozzle and the fineness of the spray desired.

The cooling effect produced varies with the weather conditions. In general, it varies from 20 to 40 deg. fahr. The final temperature is often

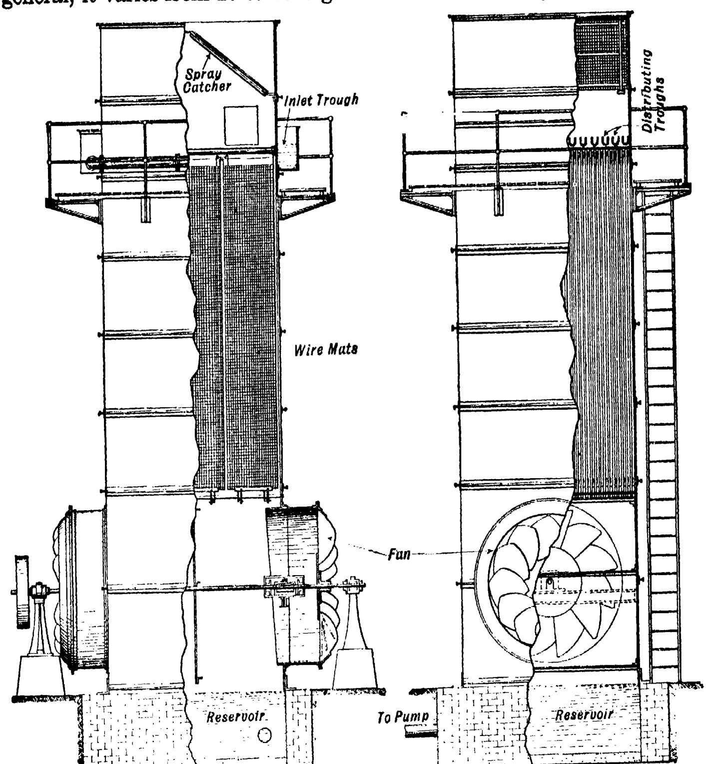


FIG. 455. — Barnard-Wheeler Water Cooling Tower.

within 3 or 4 degrees of the temperature of the air, and sometimes, under suitable conditions, may be below the temperature of the air.

536. Water-cooling Towers. — In the case of cooling towers, the water is generally delivered to the top of the tower, and is then permitted to fall to a tank below the tower. The falling water is broken into a spray, by some means that does not require pressure.

Water cooling towers may be classified as follows:

1. Forced draft.
2. Natural draft — open or atmospheric type.
3. Natural draft — closed or flue type.
4. Combined forced and natural draft.

537. Forced-draft Cooling Tower. — The *Barnard-Wheeler* water-cooling tower, shown in Fig. 455, consists of a steel tower the sides of which are entirely enclosed, except for two openings at the bottom. Within the tower are a series of removable **galvanized steel mats,** arranged in rows and hung side by side from **cross rods** located at the top of the tower. Near the top, a **steel plate open-trough** is located. From this main trough lateral branches are taken, as shown; and just above the lateral troughs, inclined wire screens are placed to catch any spray escaping. In the openings near the bottom of the tower, two fans are located.

Water from the condenser is pumped to the main distributing trough, from which it runs through the lateral troughs and is delivered in fine streams to the top of the wire mats. The water then slowly descends over the mats in a thin sheet.

Air for cooling the water is forced, by the low-pressure fans, up through the cooling tower. Opposite the discharge of each fan, deflecting plates are arranged, to direct the current of air.

538. Natural-Draft Open, or Atmospheric, Cooling Tower. — This type of tower consists of a strong wooden frame supporting the remaining parts of the tower. The walls of the tower are constructed of **louvres,** or slats, with openings between them. At the top of the tower are one or more troughs, extending the entire length of the tower, and connected to a distributing system of small V-shaped troughs, in which are cut small, triangular weirs over which the water flows. The frame and louvres are made of long-leaf yellow pine, treated to lengthen its life. Strips of cypress wood with alternate rows staggered, are used to break up the water instead of steel-wire mats used in the forced draft tower. These strips rest upon racks, each separately supported by the uprights of the main frame. Since the rows of strips are staggered, the water can drop only a few inches without being broken up, and cooling is thus aided.

The operation of this tower is much the same as that of the forced-draft tower, with the exception that the air for cooling is supplied by natural draft through the open base, and the louvres at the sides. The louvres are so constructed that the spray is prevented from escaping when the winds are heavy and when the tower is located near buildings.

539. Natural-draft, Closed or Chimney Cooling Tower. — A partial outside and sectional view of the Wheeler-Blacke closed cooling tower is shown in Fig. 456. Its construction is similar to the natural-draft open tower, with the exception that its sides are closed, and an addition, of considerable height, is erected above the portion of the tower containing the cooling surfaces and the water distribution system. Air enters at the opening at the bottom, and rises through the tower, because of the difference in temperature existing between the top of the flue and the bottom of the tower.

540. Comparisons of Cooling Towers. — The forced-draft tower requires small space and is adapted for use where (1) the space available for a tower is limited; (2) the atmospheric conditions are unfavorable; and (3) natural draft is not obtainable on account of the location of the tower. The cost of this type of tower is in excess of the open, natural-draft type.

The open, natural-draft tower requires greater space than the forced-draft tower. In general, it is adapted to about 75 per cent of all installations.

The closed, natural-draft tower is larger and more expensive than the open-type tower. The combined forced and natural draft tower is generally too expensive to have an extended application.

541. Location of Cooling Tower. — Cooling towers are located at the ground level, on a roof, or on some other elevated structure, wherever the space is available. The ground-level installation has the following advantages: (1) simplicity of foundation and reservoir construction; (2) shorter pipe lines, resulting in lower first and operating costs; and (3) localization of possible spray during high winds.

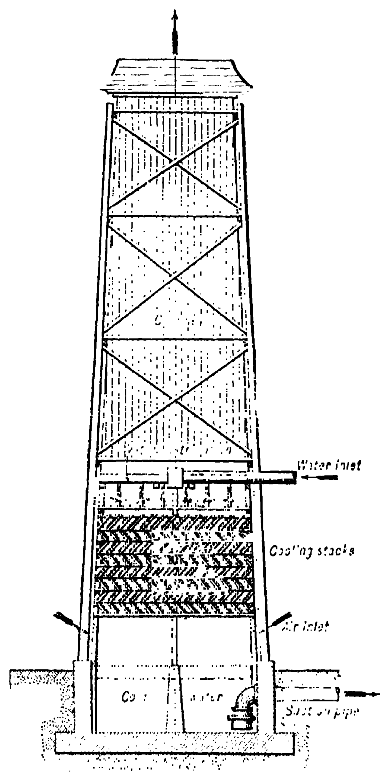


Fig. 456. — Natural Draft Cooling Tower, showing Zigzag Cooling Surface.

With cooling towers of the natural-draft type, an elevated location is often preferred, for the sake of (1) unimpeded circulation of air currents, and (2) the utilization of otherwise unoccupied space. With a surface condenser, an elevated tower is not desirable, because of increased pumping costs.

542. Principle upon which Cooling Towers Operate. — The water to be cooled gives up its heat to the rising column of air, by evaporation, convection, and radiation. Evaporation absorbs from 75 to 85 per cent of the heat; convection, or direct transfer to the air, comes next; while radiation, partly in the tower and partly through the piping, makes up the balance.

The cooling from radiation is relatively small. The amount of vapor absorbed by the air, because of evaporation, depends upon the moisture already in the air and the temperature of the air. The amount of cooling resulting from convection depends upon the temperature difference between the cooling air and the water to be cooled.

The latent heat absorbed by the cooling water, while condensing one pound of steam in the condenser, equals the quantity of heat that must be extracted in the cooling tower. The quantity of water evaporated will therefore equal the quantity condensed, less the percentage of heat removed by convection and direct radiation; that is, *the cooling tower has to evaporate a quantity of water equaling from* 75 *to* 85 *per cent of the weight of steam passing through the turbine or engine.* This loss must be replaced by a fresh supply. Expressed as percentage of the total quantity of cooling water supplied, it equals nearly 4 per cent. The water to be cooled may be lowered in temperature 40 to 50 deg. fahr.

Capacity. — The capacity of nozzles and cooling towers is stated in gallons of water cooled per minute.

REFERENCES

Spray Cooling for Industrial Purposes, Varnall Waring Co.
Cooling Water with Sprays, E. B. Badger and Sons Co.
The Spraco System, Spray Engineering Co.
Condensers and Auxiliaries, C. H. Wheeler Mfg. Co.
Steam Power Plant Engineering, Gebhardt.
Mechanical Equipment of Buildings, Vol. II, Harding and Willard.

REVIEW QUESTIONS AND PROBLEMS

1. Name the principal reason for operating engines with a condenser.

2. Classify condensers, and explain the essential differences in the fundamental types.

3. Describe the operation of (*a*) a jet condenser, (*b*) a surface condenser.

4. What type of condenser should be used to produce a high vacuum? Describe a typical condenser of this type.

7. Describe the Mullan displacement vacuum pump. Why is such a pump used?

8. Describe the operation of the ejector vacuum pump.

9. Why is it necessary to cool water used for power plant or industrial work? Name two methods used.

10. Describe the operation of the Barnard-Wheeler forced-draft cooling tower.

CHAPTER XXVI

TYPICAL MODERN POWER PLANTS

543. Foreword. — The advances made in the construction of power plant apparatus are so rapid that what is modern today will probably not be considered so a few years hence; general tendencies, however, may be discussed. The arrangement and type of equipment used in power plants depend to a large extent upon the judgment and experience of the engineers who make the plans. The tendency in recent large power stations

FIG. 457. — 60,000-kw. Cross-compound Reaction Turbine at Colfax Power Station.

is toward a small number of large units, with the equipment arranged on the unit system, each turbine having its individual boilers, condensers and piping.

In central station practice, the main units are either reaction or impulse turbines, which may have from one to three cylinders. Horizontal reaction turbines of 60,000 kilowatts capacity, having three cylinders arranged on the cross-compound principle, are being successfully used in recent plants. With this arrangement, the load is equally divided between the cylinders.

An installation of this kind, at the Colfax Plant of the Dequesne Light Co., is shown in Fig. 457. Reaction turbines of smaller capacity often have the cylinders arranged in tandem. Single cylinder units of the Curtis type, Fig. 458, having a capacity of 45,000 kilowatts, are being used in several large plants with apparent success.

The boilers supplying steam to these large-capacity turbines have been correspondingly increased in capacity, and are either of the water-tube

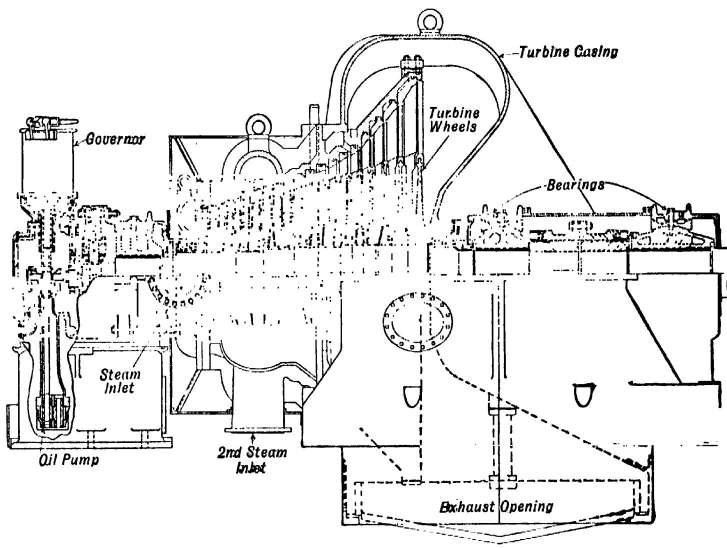


FIG. 458. — Cross-Section of Curtis 45,000-kw. Turbine.

cross-drum or Stirling type, having as high as 26,000 square feet of heating surface in a single boiler. This amount of surface is obtained by using a large number of long tubes. The drums of these boilers are set high, to give large furnace volume and thus permit running at high ratings; for example, the distance from the floor to the nozzles, of the boilers installed in the Highland Park Plant of the Ford Motor Car Co., is 35 feet 8 inches. The boilers used in the new Delaware Power Station of the Philadelphia Electric Company are of the Stirling type, with 15,080 square feet of heating surface, exclusive of the internal economizer, and those in the Hartford Electric Co. are Biglow Hornsby type water-tube boilers, having a heating surface of 13,920 square feet. In large cities, where land has a high value, the boilers are arranged on two or more floors, one above the other, with the coal bunkers above the upper boiler-room floor.

The auxiliaries serving these large-capacity main units have also been increased in capacity and are driven either by motors or by a combination

of motors and turbines. When they are motor-driven at constant speed, alternating-current motors are used, with a voltage of 2300 for motors above 75 horsepower and 440 for smaller motors. With auxiliaries requiring variable speed, direct-current motors are used, at a voltage of 250. The current for the motors may be taken from the main turbine, or may be supplied, as in recent installations, by a separate turbine known as the **house turbine.**

Surface condensers having 56,000 square feet or more, of condensing surface in a single shell are now in operation. In some cases the condensing surface used to serve a large main unit is made in several shells of about 20,000 square feet each. Where the condensing water is taken from rivers in which the water level fluctuates widely, the condensing equipment is located in waterproof concrete wells, to obviate the pumping of large quantities of condensing water through high heads. In some cases these wells are 75 feet deep and 50 feet in diameter. The pipe connecting the condenser and turbine, in this case, is large and long, and a special type of expansion joint is used at the turbine end of the exhaust pipe. Jet condensers capable of handling 13,000,000 gallons of water per hour are giving satisfaction.

In the larger plants, turbine- or motor-driven centrifugal pumps are used instead of direct-acting steam pumps, in sizes ranging from twenty up to several hundred horsepower per unit. These pumps are used for boiler feeding, circulating condensing water, and in some cases for fire service.

The stokers which serve the large boilers are, in the majority of installations, of the multi-retort underfeed type.

The increased overloads carried on boilers at the present time have made it necessary to provide a dependable means of eliminating the impurities in boiler feedwater. In several of the large plants, evaporators of either the high-pressure or the vacuum type are being used for this purpose, and are apparently giving satisfactory results. In the majority of the central station plants, the intermittent soda and lime process of treating feedwater is used.

In plants using economizers with steel tubes, pitting at low water temperatures is prevented by removing the air from the condensate. This is accomplished by an air extractor, or de-aërator, which boils the feedwater by maintaining a vacuum in a tank through which the feedwater passes.

In some plants the air for combustion is preheated by utilizing the heat in the furnace gases.

To illustrate the tendencies in various types of steam power plants, the following will be briefly described:

1. Central Station, Calumet Station of Commonwealth Edison Company.
2. Isolated Plant, Dodge Bros., Motor Car Company.
3. Steam Plant using Powdered Coal.

4. Steam Plant with Oil-fired Boilers.
5. Marine Steam Power Plant.

544. Central Station, Calumet Power Station.—The Calumet power station of the Commonwealth Edison Co., Chicago, will eventually be one of the largest stations in the United States. Its ultimate capacity will be at least 240,000 kilowatts, of which amount 60,000 kilowatts is now installed with 60,000 kilowatts under construction. The steam pressure, at the boiler, is 330 pounds per square inch gage and, at the turbine, 300 pounds with a superheat of 200 deg. fahr. thus giving a final temperature of 622 deg. fahr.

The arrangement of the equipment is shown in Fig. 459, by a sectional elevational, and in Fig. 460, by a plan view of the station. The boilers are located on the second floor, and are arranged so that three boilers supply steam to one turbine unit. Each boiler is a cross-drum B. and W. boiler with a steam-making surface of 15,089 square feet, having an integral superheater of 4,052 square feet superheating surface. The baffles are horizontal, with the lower row of tubes placed below the lower baffles. The front tube header is 15 feet 4 inches above the floor, thus giving a large furnace volume per square feet of heating surface. Just above the boiler, a steel-tube economizer, having a heating surface of 9,669 square feet, is arranged in three sections, with the tubes running at right angles to the boiler tubes.

The boilers are served by **forced-draft chain-grate stokers** set under the low end; two stokers, having a total of 364 square feet of grate surface, are used under each of six boilers, and a single large stoker, 24 feet wide by 18 feet 6 inches long, under one boiler. The stokers are driven by direct-current motors, so that the rate of feed may be changed, and have a flat suspended arch with the new radial end. The bridge wall is eliminated, and the stokers extend back to the rear wall, which is protected by water backs. The side and center walls are protected at the clinker line by ventilated tile. The air for combustion is drawn, from the boiler room or train shed, by induction motor-driven multi-vane fans located on the boiler room floor back of their respective boilers. This air is supplied to the stokers through ducts arranged in six zones; with an 8-inch fire, the pressure in the first zone is 0.8 inches water, and in the sixth zone 0.2 inches water. The gases are removed from the furnace by induced draft fans located at either side of the boiler house and on a level with the economizer. These fans discharge upward into a smoke flue, on top of the roof, leading to a centrally located self-supporting steel stack, 200 feet high and 16 feet inside diameter.

Coal is delivered in cars, which enter 32 feet below the central firing aisle, and which are unloaded, by a traveling crane, either into a large storage pit or into a traveling hopper with feeder located on the side of the train shed. From this hopper the coal is fed to a belt conveyor of 400 tons per hour

capacity, and is then delivered to a coal breaker. Thence it passes through collecting hoppers to a short belt conveyor feeding into two pivoted bucket conveyors, which elevate the coal to the bunker level and discharge it into

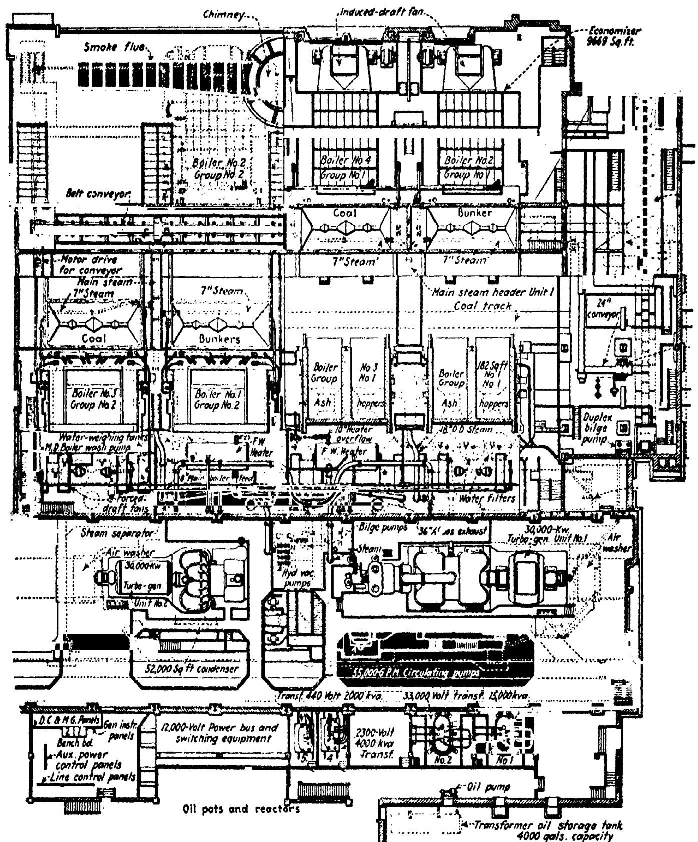


FIG. 460. — Plan View of Calumet Steam Power Station.

receiving hoppers. It is then discharged directly to belt conveyors running over reinforced concrete bunkers of 250 tons capacity per boiler. From the bunkers, coal is delivered to the stokers through sheet-steel spouts. The ashes drop from the chain grates into ash hoppers, which discharge directly into railway cars.

The high-pressure steam piping is made of steel, double-extra heavy with steel fittings and welded joints. Two 7-inch leads rise from each boiler and

are united by a special Y-fitting to form a single line leading to the 18-inch main header for each turbine unit.

Pitting of the steel economizer tubes is prevented by maintaining the feedwater temperature at the economizer at 175 deg. fahr., by means of a closed feedwater heater located between the condenser hot well and the open feedwater heater. *To deliver the water to the open heater at the proper*

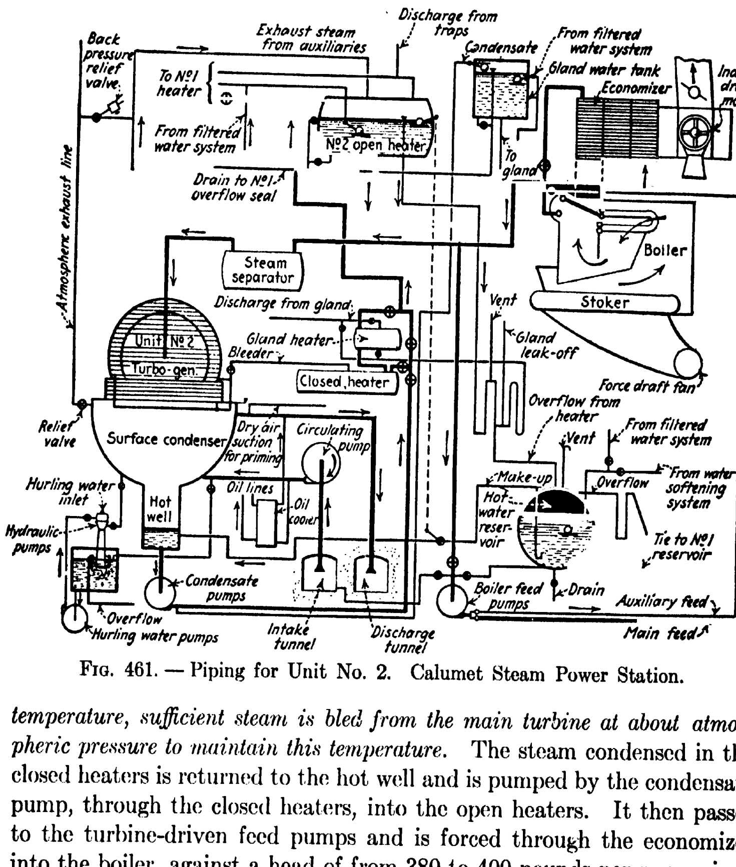


FIG. 461. — Piping for Unit No. 2. Calumet Steam Power Station.

temperature, sufficient steam is bled from the main turbine at about atmospheric pressure to maintain this temperature. The steam condensed in the closed heaters is returned to the hot well and is pumped by the condensate pump, through the closed heaters, into the open heaters. It then passes to the turbine-driven feed pumps and is forced through the economizer into the boiler, against a head of from 380 to 400 pounds per square inch gage. The primary water supply is the condenser hot well; and the make-up water, which amounts to about 5 per cent, is supplied by a 4000 gallon per hour Permutit water-softening system.

The main units are two in number, each having a capacity of 30,000 kilowatts. Unit No. 1 is a tandem-compound machine with a speed of 1200 r.p.m., and unit No. 2 is a single-cylinder machine operating at 1800 r.p.m. The condensers for these machines are of the surface type, with

a condensing surface of 52,000 square feet each. They are capable of maintaining a vacuum of 28.5 inches to 29 inches mercury, with cooling water ranging from 45 deg. to 70 deg. fahr. and can condense 33,000 pounds of steam per hour. The circulating and condensate pumps are installed in duplicate, one being driven by a steam turbine, the other by an induction motor. The air pumps for unit No. 1 are of the Leblanc type, and those for unit No. 2 are hydraulic-type air pumps. The arrangement of the piping and auxiliaries is shown in Fig. 461 for unit No. 2.

The condenser circulating water is taken from the Calumet River through two **open flumes** having **revolving** and **stationary screens** at the cribhouse, and is discharged to the river through another open flume.

The air for cooling the generator is kept cool and clean by being passed through an **air washer.** The lubricating oil is not filtered, but when dirty is collected in a 2000-gallon tank, and once a week is passed through a De Laval centrifugal separator to remove the moisture and dirt.

The size and type of equipment in this plant are tabulated in Table 36.

545. Isolated Steam Power Plant, Dodge Bros. Motor Car Co. — The modern isolated plant differs from the central station plant mainly in the size of the units installed. The plant of the Dodge Bros., Fig. 462, well illustrates this. It was designed for reliable and continuous service and is arranged on the unit system; that is, a turbo-generator with its boiler and auxiliaries comprise a plant unit capable in itself of being operated independently of any other unit. The ultimate capacity of the plant is 50,000 kilovolt-amperes with an average **load factor;** *that is, the ratio of the average load to the maximum load during a certain period of time,* of 55 per cent. The operating steam pressure is 225 pounds per square inch gage, with a superheat of 100 deg. fahr.

The boilers are located on a floor 35 feet above grade and are of the Stirling type, having a heating surface of 12,870 square feet. Each boiler is capable of evaporating 89,000 pounds of steam per hour at 200 per cent of rating. Two boilers form a unit which is served by an underfeed stoker with an individual motor drive, a forced-draft fan and a boiler feed pump per unit. Between the first and second banks of tubes, sufficient heating surface has been provided to give a superheat of 100 degrees, and space has been provided for economizers and induced-draft fans, should it be decided to install them later.

An underfeed Riley stoker, having twelve extra long retorts, is used for each boiler, and is driven by a 15-horsepower compound-wound direct-current motor through a silent chain. The air for combustion is supplied to each boiler by a multi-vane forced draft fan capable of delivering 54,200 cubic feet of air per minute at a static pressure of $7\frac{1}{4}$ inches and located on a floor 13 feet below the boiler room floor. The products of combustion pass through overhead smoke flues with easy bends to op-

TABLE 36.—PRINCIPAL EQUIPMENT OF CALUMET STATION, COMMONWEALTH EDISON CO., CHICAGO

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
7	Boilers	Cross-drum	15,089 sq. ft., H.S.	Steam generator	325 lb. press.	Babcock and Wilcox Co.
7	Superheaters	Integral	4,052 sq. ft., H.S.	Superheating steam	200 deg. rise in temperature	Babcock and Wilcox Co.
12	Stokers	Chain grate	182 sq. ft., G. S.	Boiler furnaces	Forced draft, d.c. motor drive	Combustion Eng. Corp.
1	Stoker	Chain grate	444 sq. ft., G.S.	Boiler furnaces	Forced draft, d.c. motor drive	Combustion Eng. Corp.
7	Economizers	Steel tubes and headers	9,669 sq. ft., H.S.	Heating feedwater	Receives water at 175° F.	Babcock and Wilcox Co.
2	Fans	Squirrel cage	153,000 cu. ft. per min.	Induced draft	200 hp. 2300 volt, motor driven	B. F. Sturtevant Co.
7	Fans	Multi-vane	40,000 to 60,000 cu. ft. per min.	Forced draft	50 hp. 440 volt, motor driven	B. F. Sturtevant Co.
2	Chimneys	Self-supporting steel	200 ft. high, 16 ft. diam.	Discharge gases	4-boilers per chimney	Amer. Chimney Const. Co.
1	Hopper	Traveling	10 × 13 ft.	Coal handling	Motor driven, 10 hp., 440 volt	Robins Conveying Belt Co.
1	Conveyor	Belt	400 tons per hour	Coal handling	Motor driven, 30 hp., 440 volt	Robins Conveying Belt Co.
5	Conveyors	Belt	200 tons per hour	Coal handling	Motor driven, 30 hp., 440 volt	Robins Conveying Belt Co.
1	Conveyor	Pivoted bucket	200 tons per hour	Coal handling	Motor driven, 30 hp., 440 volt	Mead Morrison Co.
1	Breaker	Bradford	12 ft. diam. × 17 ft. long	Breaking coal	Motor driven, 30 hp., 440 volt	Penn. Crusher Co.
1	Crusher	Two-roll	30 × 30 in.	Breaking coal	Motor driven, 30 hp., 440 volt	Robins Conveying Belt Co.
1	Crane	Electric bridge	175 tons per hr.	Coal from car to belt	Motor driven, 20 hp., 440 volt	Whiting Corporation
1	Turbo-generator	Tandem-compound	30,000 kw.	Electric supply	300 lb. press., 200° super., 1200 r.p.m.	Westinghouse Mfg. Co.
1	Turbo-generator	17 stage-Curtis	30,000 kw.	Electric supply	300 lb. press., 200° super., 1800 r.p.m.	General Elect. Co.
1	Condenser	Surface	52,000 sq. ft.	Serves turbine	28.5 to 29 in. vacuum	Westinghouse Mfg. Co.
2	Pumps	2-impeller centrifugal	55,000 gal. per min.	Circulating water	800 hp. motor driven	....
2	Pump	Le Blanc entrainment	....	Air pump	100 hp. turbine, 150 hp. motor	Westinghouse Mfg. Co.
1	Condenser	Surface	52,000 sq. ft.	Serves G.E. turbine	28.5 to 29 in. vacuum	Worthington Pump & Mach. Co.
1	Pump	1-impeller centrifugal	55,000 gal. per min.	Circulating water	800 hp. motor driven	....
4	Pumps	Centrifugal	1000 gal. per min.	Condensate	Turbine and motor driven	....
2	Pumps	Hurling water	....	Air pump	....	....
2	Heaters	Open	400,000 lb. per hour	Feedwater	....	Warren Webster & Co.
2	Heaters	Closed	400,000 lb. per hour	Feedwater	....	Patterson-Kelly Co.
3	Pumps	4-stage Jeanesville	900 gal. per min.	Boiler feed	389 hp. turbine driven	Worthington Pump & Mach. Co.
1	Air compressor	Compound	625 cu. ft. per min.	....	150 hp. motor driven	Sullivan Mach. Co.

Miscellaneous Equipment: Permutit water softening system; "s-c" feedwater regulators; Republic and Bailey steam flow meters; Ellison draft gages; Taylor recording thermometers; 7-Diamond soot blowers; 3-water filters; 2-Carrier air conditioners.

posite sides of a self-supporting steel stack, which is supported on the central columns of the building structure. The stack is 235 feet high above the boiler room floor and has an inside diameter of 13 feet. The flue opening at the stack has an area of 95 square feet.

Each boiler is served by a fourteen-element soot-blower, and the blow-off valves are located conveniently in the offset back of the bridge wall.

The ashes are discharged by the stoker into large ash hoppers, which have outlets passing through the fan room floor, so that the ashes may be discharged directly into railway cars. The coal comes in on two tracks and is dumped into an unloading hopper, from which it is transferred by apron feeders to four-roll crushers delivering to 30-inch belt conveyors rising on an incline to two pivoted bucket conveyors. The latter elevate the coal and discharge it to either of two 175-foot, 24-inch belt conveyors running over the bunkers. Coal from the bunkers is delivered to two stationary scales, which discharge into structural steel spouts for delivery to the stoker hopper. All of the conveyors are driven by induction motors.

The feedwater is collected in three large tanks located at the grade level, and falls by gravity to either a motor-driven hot-well pump or a turbine-driven pump, and is pumped to the two feedwater heaters, each having a capacity of 300,000 pounds per hour. The heaters are located on a balcony above the boiler room floor. The feedwater pumps, of which there are four — three electrically driven by a 175-horsepower direct-current motor, and one turbine-driven — are located on the boiler room floor and take their suction from either of two 8-inch headers leading to the feedwater heaters. The electrically driven pumps have two four-stage compartments arranged in series and capable of pumping against a head of 720 feet at 1200 r.p.m. The discharge from each pump branches into two 6-inch lines which cross the boiler room and connect into two headers, from each of which there is a connection to each boiler. A Venturi meter is located in each line, and between the two lines there are two cross-connections, one containing a feedwater regulator and the other an open line, so that the water may be fed automatically or by hand. *The make-up water will eventually be supplied by an evaporator.*

The main steam piping is made of steel with bends to allow for expansion. It is located in the fan room with leads rising to the turbines, which, at present, are two reaction-type 4000-kilowatt units, generating current at 3-phase 60 cycles and operating at 200 pounds per square inch pressure, 100 deg. fahr. superheat and a 28 inch vacuum. The turbines are supported by a heavy structural steel framework, from which the surface condenser is supported by lugs. The condenser auxiliaries are driven by direct-current motors. The circulating pump is an 18-in. centrifugal pump; the condensate pump a 4-inch; and the air pump an evactor type pump with a two-stage steam jet, intercooler and surface-

TABLE 37.—EQUIPMENT OF DODGE BROTHERS POWER PLANT

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
6	Boilers	Water-tube	12,870 sq. ft.	Generate steam	225 lb., 100°, superheat, forced draft	D. Connelly Boiler Co.
6	Stokers	Underfeed	12-retort, extra long	Serve boilers	Driven by 20-hp. G.E. motor, d.c., 200–1,200 r.p.m	Sanford Riley Stoker Co.
6	Fans	Turbo-vane	54,200 c.f.m.	Forced draft	Driven by 100-hp, d.c., G.E. motor, 325–950 r.p.m.	B. F. Sturtevant Co.
6	Superheaters	Foster		Superheat steam	100° F. superheat	Power Specialty Co.
3	Pumps	8-stage centrifugal	5-in., 450 g.p.m.	Boiler feed	Driven by 175-hp., d.c., G.E. motor, 1,100 r.p.m.	De Laval Steam Turbine Co.
1	Pumps	4-stage centrifugal	600 g.p.m.	Boiler feed	Driven by De Laval 200 hp. turbine, 2,400 r.p.m.	De Laval Steam Turbine Co.
2	Heaters	Open	360,000 lb. per hr., 100–220° F.	Feed water	Exhaust steam	Harrison Safety Boiler Co.
2	Conveyors	Apron feeders	30 in. wide; 97 and 104 ft. long	Coal from truck hoppers to crushers	Driven by 10-hp. G.E. ind. motor, 1,200 r.p.m.	Link Belt Co.
2	Crushers	Four-roll	24 × 36 in., 75 tons per hr.	Crush coal	Driven by 40-hp. G.E. ind. motor, 1,200 r.p.m.	Link Belt Co.
2	Conveyors	Belt	30-in. wide, 63 and 78 ft. long	Coal from crushers to bucket conv.	Driven by 5-hp. G.E. ind. motor, 1,200 r.p.m	Link Belt Co.
2	Conveyors	Peck carriers	24 × 30 in., 105 ft. centers	Elevate coal to bunker level	Driven by 15-hp. G.E. ind. motor, 1,200 r.p.m.	Link Belt Co.
2	Conveyors	Belt	24-in. wide, 175 ft. long	Distribute coal to bunkers	Driven by 10-hp. G.E. ind. motor, 1,200 r.p.m.	Link Belt Co.
3	Pumps	Centrifugal	4-in.	Hotwell	Driven by 40-hp. G.E. ind. motor, 1,200 r.p.m.	Allis-Chalmers Co.
2	Pumps	Centrifugal	2-in.	House Service	Driven by 7½-hp. G.E. ind. motor, 1,750 r.p.m.	Allis-Chalmers Co.
2	Turbines	Cond. non-cond.	5,000 kva.	Generating units	200 lb. press., 100° superheat, 3,600 r.p.m.	Allis-Chalmers Co.
2	Generators	A.C.	5,000 kva.	Generating units	3-phase, 60-cycle 4,600-V., 3,600 r.p.m.	Allis-Chalmers Co.
2	Condensers	Surface	9,500 sq. ft.	Serve turbines	28-in. vacuum	Allis-Chalmers Co.
2	Pumps	Centrifugal	18-in.	Condenser circulating	Driven by 200-hp. G.E., d.c. motor, 600 r.p.m.	Allis-Chalmers Co.
2	Pumps	Centrifugal	4-in.	Condenser condensate	Driven by 10-hp. G.E., d.c. motor, 1,150 r.p.m.	Allis-Chalmers Co.
2	Evactors	Two-stage with after-cooler		Condenser air pump		Allis-Chalmers Co.
1	Pump	Centrifugal	4-in.	Emergency	Driven by Moore 40-hp. turbine, 1,750 r.p.m.	Allis-Chalmers Co.
1	Crane	Electric bridge	50-ton, 4-motor	Turbine room	58-ft. span	Northern Engin'g Co.
1	Cooling tower	Natural and induced draft	9 element, 18,000 g.p.m.	Condenser circ. water	105° to 85°, temp. 72°, humidity 69 per cent	Wheeler Cond. & Eng. Co.
18	Fans	Disk	120 in.	Serve cooling tower	Driven by 25-hp. G.E. ind. motor, 900 r.p.m.	Wheeler Cond. & Eng. Co.

Vulcan soot blowers, 14 element; Copes feed-water regulators; Schutte & Keorting stop check valves; Yarway blow-off valves; Cranetilt traps; Allis-Chalmers 100-kw. exciters; 4,600 volt, switches and control, Westinghouse; Insulators and fittings, Gen'l Device and Fittings Co.; 440-volt control, Industrial Controller Co.; 220-volt d.c. control, General Electric.

type condenser. The condensate from the main condenser is forced through this after-condenser before passing to the hot-well tanks.

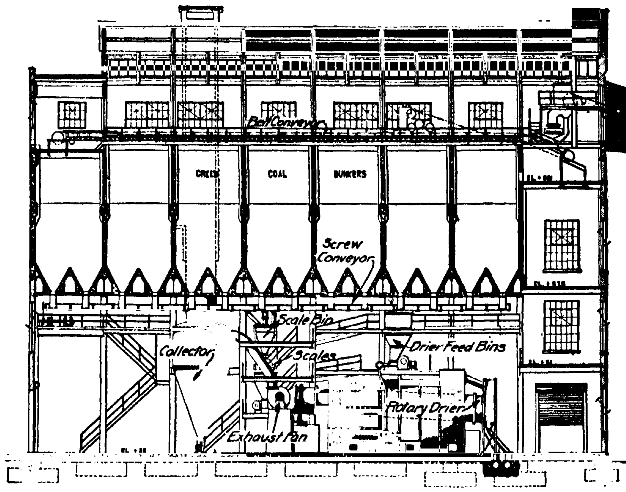
A cooling tower of the combined natural and forced draft type is used to cool the condensing water, and is capable of cooling 25,000 gallons per minute. It is the largest of its type and is located on top of a loading platform, 33 feet above grade level. (See Table **37** for list of equipment).

546. Pulverized-fuel-burning Power Plant. — The Lakeside Power Station of the Milwaukee Electric Railway and Light Co., is the largest pulverized-fuel-burning plant in the world, with an ultimate capacity of 200,000 kilowatt, of which 40,000 kilowatt is now installed. The plant is arranged in four distinct sections, namely, the pulverizing room, the boiler room, the turbine room, and the switch house. The first three sections will be briefly described.

Coal is delivered by rail to a car-dumping and crushing plant located on the shore of Lake Michigan, about 400 feet from the plant. The coal cars are dumped by a revolving dumper into a track hopper fitted with a bottom shaker feeder. Belt conveyors carry the coal from the track hopper over a magnetic pulley, which takes out the tramp iron, before the coal is discharged to a two-roll crusher and hammer mill. The crushed coal is received by a belt 36 inches wide and 385 feet long, and is elevated 91 feet to distributing belts over the green coal bunkers in the pulverizing plant, Fig. 463.

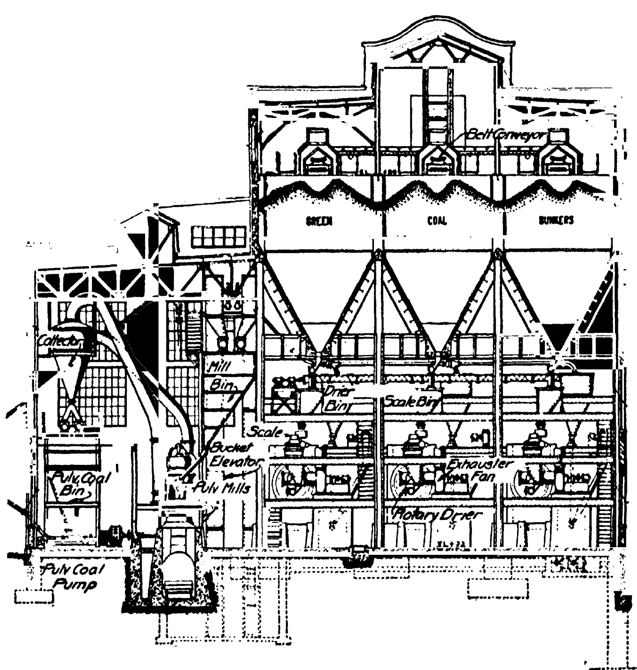
Green coal from the bunkers, which hold 3400 tons, is carried by three screw conveyors to automatic weighing scales and thence by another set of screw conveyors to the indirect-fired dryers which reduce the moisture from 10 to 2 per cent. Screw conveyors take the coal from the driers to bucket elevators, which in turn discharge to other screw conveyors for delivery into the dried coal bins, located over the eight air-separating unit-pulverizing mills. These mills reduce the coal in size, so that 75 per cent will pass through a 200-mesh screen and 90 per cent through a 100-mesh screen. A fan carries this fine powder to a cyclone separator, where the fuel is separated from the air and falls by gravity to a screw conveyor which conveys it to the pulverized fuel bins. The fuel is delivered from these fuel bins to those over the boilers, by an air-conveying system.

The boiler room is arranged as shown in Fig. 464. The fuel is fed from the pulverized fuel bins, by screw conveyors, to burners which inject the fuel downward, just inside the front wall of the furnace, the bottom of which is cooled by water grates in order to prevent slag. An economizer, equipped with an induced draft fan, serves each boiler and raises the temperature of the feedwater from 140 deg. to 255 deg. fahr. The flues leading from the boiler are made of concrete, are located beneath the economizer, and discharge into two reinforced concrete chimneys located between the pulverizing room and the boiler room. The chimneys are 220 feet high, with an inside diameter at the top of 15 feet.



LONGITUDINAL SECTION OF PULVERIZING PLANT

FIG. 463*a*. — Pulverizing Plant.



Pulverizing Plant

FIG. 463*b*. — Pulverizing Plant.

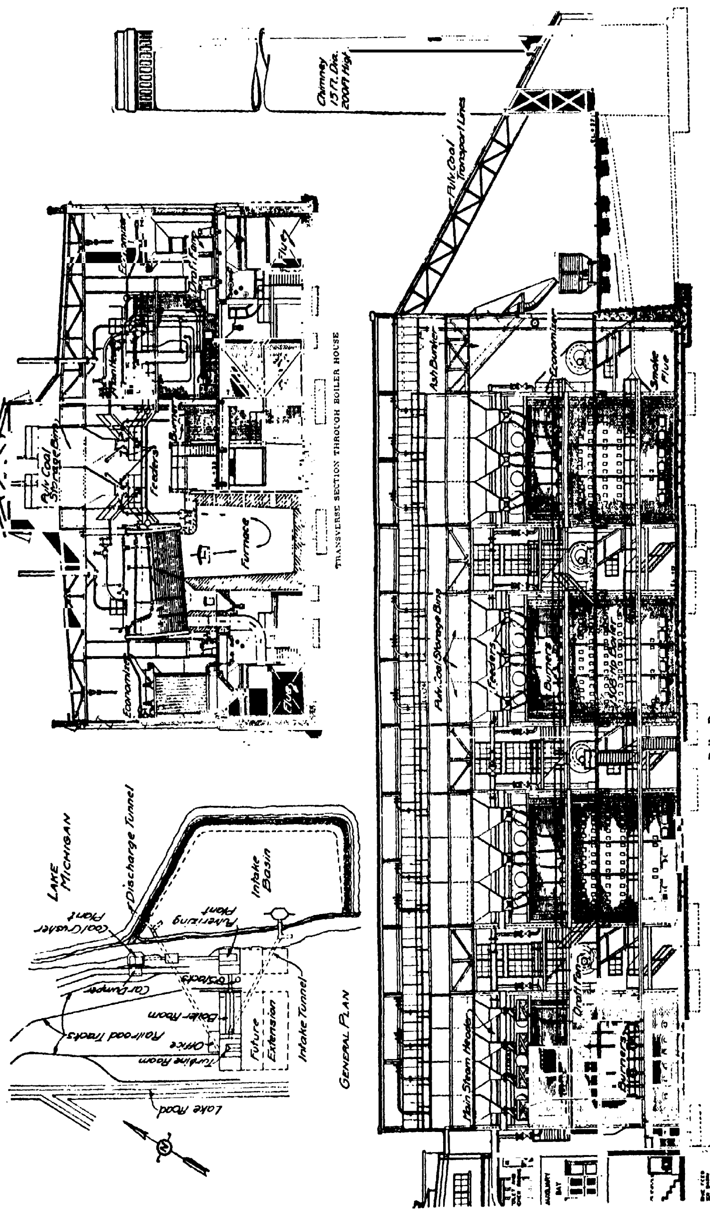


FIG. 464. — General Layout of Boiler Room — Milwaukee Electric Light & Power Co.

TABLE 38.—EQUIPMENT OF THE LAKESIDE STATION, MILWAUKEE ELECTRIC RAILWAY AND LIGHT CO.

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
1	Conveyor	Belt	36 in. wide, 250 tons per hr.	Coal from hopper to crusher	250 tons per hr. at 250 ft. per min.	Robins Conveying Belt Co.
1	Crusher	Two-roll	150 tons per hr.			Robins Conveying Belt Co.
1	Conveyor	Belt	250 tons per hr.	Delivers coal to ?		Robins Conveying Belt Co.
3	Conveyors	Belt		Distributes coal to bunkers		Robins Conveying Belt Co.
3	Conveyors	Screw		Coal from green coal bunker to scales		Robins Conveying Belt Co.
3	Scales	Automatic weighing				
3	Dryers	Horizontal	10 tons per hour	Drying coal	Reduces moisture from 10 to 2 per cent	Fuller Lehigh Co.
8	Pulverizing mills	Air separating	46-in. 6 tons per hr.	Pulverizing coal	75 per cent through a 200 mesh screen	Fuller Lehigh Co.
8	Motors		100-hp.	Driving mills		Fuller Lehigh Co.
8	Fans			Lifts pulverized fuel		
	Conveyor	Screw		Pulverized material		
8	Boilers	Water-tube	1,306 hp.	Steam generator	255 lb. press., 250 per cent rating	Edgemore Boiler Co.
8	Superheaters	Attached	90,100 lb. per hr.		Superheats from 411 to 611° F.	Foster Superheater Co.
8	Economizers				Raises temperature from 140 to 255° F.	B. F. Sturtevant Co.
8	Fans			Induced draft	Turbine driven	B. F. Sturtevant Co.
2	Stacks	Concrete	220 ft. high, 16 ft. 6 in. diam.	Remove products of combustion		
	Heaters	Horizontal open		Heat feedwater	Temp. from 80 to 140° F.	Hoppes Mfg. Co.
2	Meters	Lea V-notch	800,000 lb. per hr.	Measure feedwater	Located 15 ft. above pumps	Lea Courtenay Co.
2	Pumps	Centrifugal	6in.-650 r.p.m.	Boiler feed	250 hp. steam turbine driven	Lea Courtenay Co.
2	Pumps	Centrifugal	4 in.-450 r.p.m.	Boiler feed	150 hp. motor driven	Lea Courtenay Co.
1	Conveyor	Steam jet		Ash conveyor		
2	Turbo-generators	Curtis	20,000 kw., 3 phase	Generating current	250 lb. press., 200° superheat 1 lb. abs. back pressure	General Electric Co.
2	Condensers	3-pass surface	35,000 sq. ft.	Condensing steam		Wheeler Condenser Co.
2	Pumps	Centrifugal	24-in., 18,000 g.p.m.	Circulating water	170 hp. motor driven	Wheeler Condenser Co.
2	Pumps	Centrifugal	24-in., 18,000 g.p.m.	Circulating water	170 hp. motor driven	Wheeler Condenser Co.
2	Pumps	Centrifugal		Condensate	50 hp. motor driven	Wheeler Condenser Co.
	Pump	Turbo		Air removal	100 hp. motor driven	Wheeler Condenser Co.
	Pump	Steam jet		Air removal		Wheeler Condenser Co.

Miscellaneous Equipment: 1 Revolving car dumper; 1 Two roll crusher; 1 Hammer mill; Non-suspension coal bunkers; Bucket elevators; cyclone separator; Pulverized fuel bins; 32-Locupolo burners; 32 Screw feeders; Flue gas recorder; 2-Air washers; 1-air compressor.

The feedwater consists mainly of returns from the condensers and is pumped to an overhead hot-well tank, from which it feeds by gravity to two Hoppes open heaters. Leaving the heater, the water flows over a V-notch weir to the four feed pumps, two of which are driven by steam turbines and two by induction motors.

The ash accumulation is removed by a steam-jet ash conveyor, and is delivered to an ash bunker which discharges into cars on tracks located at the end of the boiler house.

The turbine room has an elevation of 33 feet above grade level, and contains two 20,000-kilowatt turbo-generators, each of which is served by a 35,000 square foot surface condenser, located 32 feet below the turbine room floor in order to minimize the " head " against which the two 24-inch circulating pumps work. The cooling water is taken from a **rubble mound enclosure** which extends about 485 feet out into Lake Michigan. An intake tunnel carries the water from the lake to an intake chamber below the condensers. The condensate is removed from the condenser by a centrifugal pump, and the air by a turbo-air pump, with a steam-jet air pump held in reserve.

A house turbine is provided for station use, and an air washer for each generator. The latter draws its air supply from the main generator room, and the heated air is discharged through ducts into the basement of the boiler room, in order to augment the regular air supply. The ducts are arranged to by-pass sufficient air back into the turbine room to heat it during cold weather. High-pressure steam can be admitted to the generator to extinguish fire.

The economy of this station, when completed and operated at full-load factor, will be that resulting from a combined boiler, furnace, superheater, and economizer efficiency of 88.5 per cent, using Illinois coal with a heat value, as fired, of approximately 11,000 B.t.u.

The main equipment installed in the station is given in Table 38.

547. High-pressure Oil-burning Installation. — The boiler plant of the Fall River Electric Light Co. is a typical installation. The equipment consists of Babcock and Wilcox Lodi-type mechanical burners, Fig. 465, for atomizing the oil and mixing it with the air required for combustion; Blake and Knowles duplex vertical-type pumps capable of exerting a pressure of 300 pounds per square inch, for pumping the oil from the tanks and delivering it to the burners; Whitlock coil heaters with steel shells good for 200 pounds per square inch pressure, for raising the temperature of the oil to the point required for atomizing; and the Merit automatic-control system, described in *Power*, Oct. 4, 1921, for the regulation of the dampers and oil atomizers. This equipment, together with the service tanks and suction pipe lines is installed in duplicate to ensure continuity of service. The oil feed and return lines, Fig. 466, form a ring system.

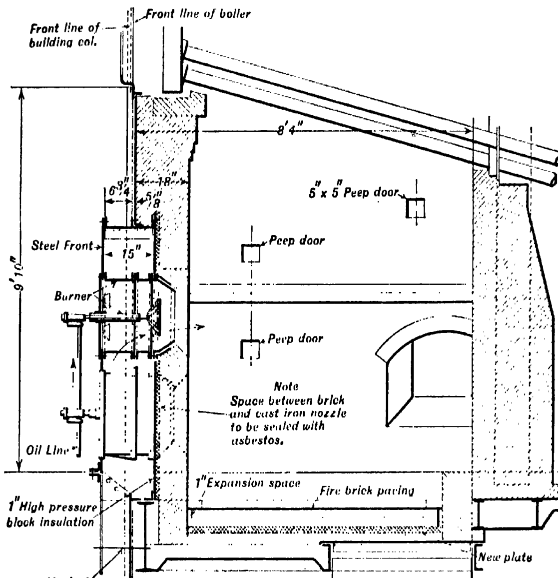


Fig. 465. — Boiler Furnace with Lodi Mechanical Atomizing Burner.

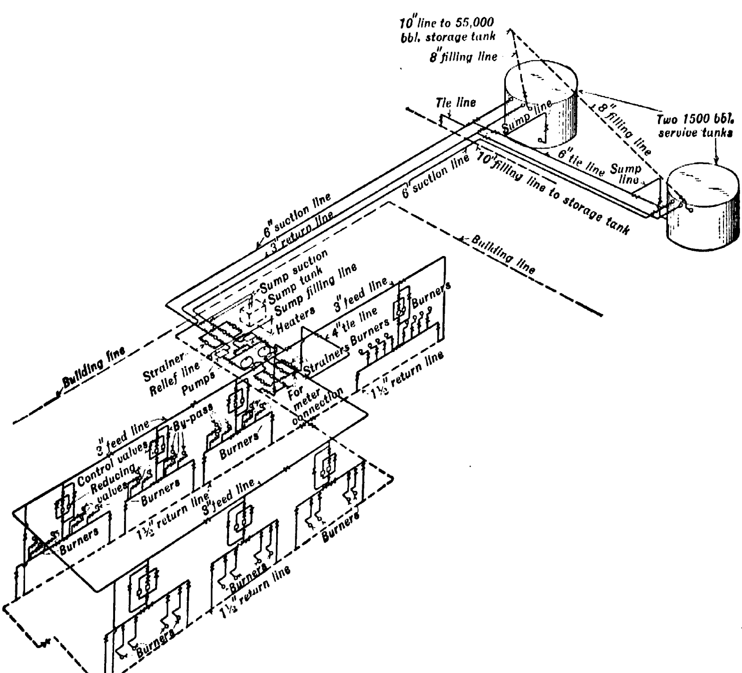


Fig. 466. — Isometric View of Oil Piping at the Fall River Electric Light Company.

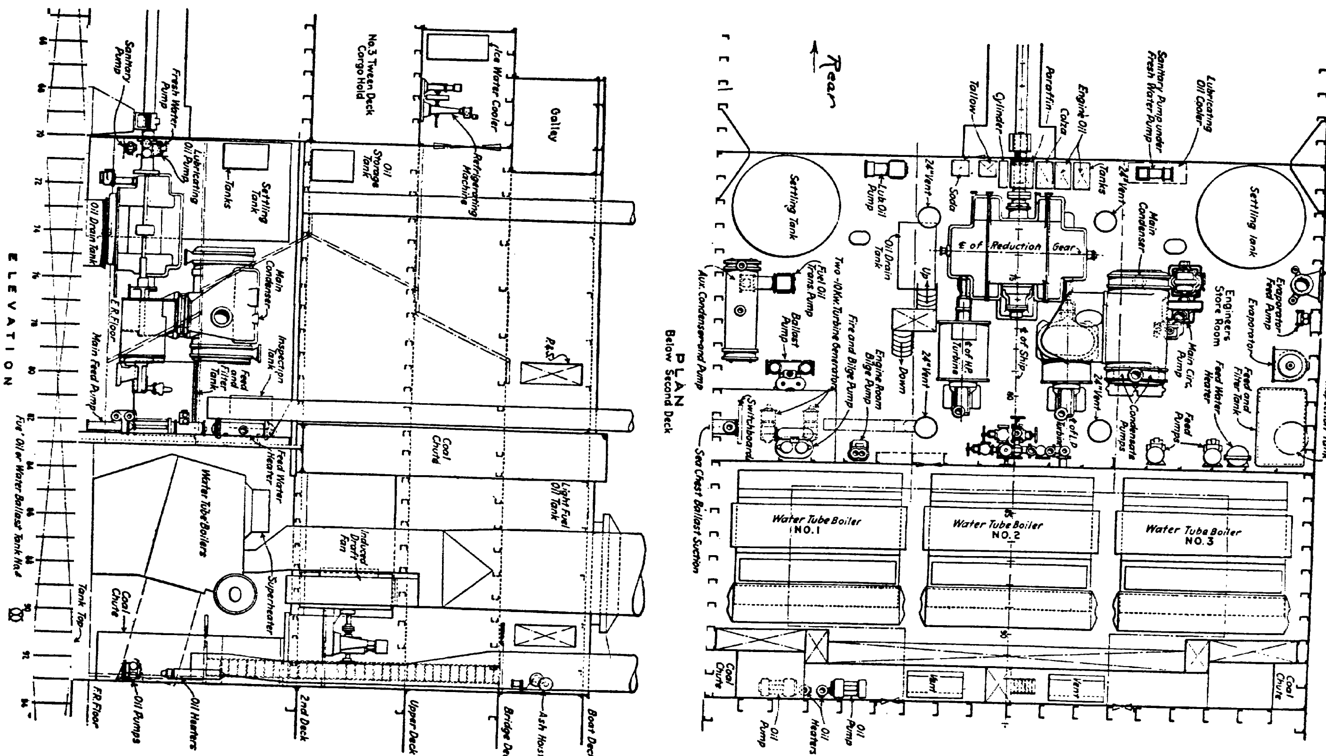


FIG. 467. — Arrangement of Equipment in a Steam Marine Power Plant — Cargo Vessel.

TABLE 39. — PRINCIPAL EQUIPMENT OF A 9000-TON DEAD WEIGHT CAPACITY "MERCHANT" STEAM DRIVEN SHIP

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
3	Boilers	Water-tube, cross-drum	3025 sq. ft., H. S.	Steam generator	200 lb. press., induced draft	Babcock & Wilcox Co.
2	Fans	Plate	90 in. diameter	Induced draft	Engine driven — 22,000 cu. ft. per min.	
2	Pumps	Vertical simplex	12 × 8 × 24 in.	Feedwater	200 lb. press., 15 strokes per min.	Nat. Transit P. & M. Co.
1	Pump	Vertical duplex	6 × 5¾ × 6 in.	Bilge pump	Operates on reduced steam pressure	Nat. Transit P. & M. Co.
1	Pump	Hor. duplex	7¼ × 7½ × 10 in.	Fuel oil transfer		Nat. Transit P. & M. Co.
1	Pump	Vertical duplex	10 × 8 × 12 in.	Fire and bilge		Nat. Transit P. & M. Co.
1	Pump	Vertical duplex	4½ × 4 × 4 in.	Evaporater feed pump		Nat. Transit P. & M. Co.
2	Pumps	Hor. duplex	5¼ × 3½ × 5 in.	Fuel oil service		Nat. Transit P. & M. Co.
1	Hoist			Removal of ashes	Hand operated	
1	Evaporator		50,000 lb. per hr.	Supply pure water		
1	Turbine	Compound reaction	2,500 shaft hp.	Drives propeller	200 lb. press., 28½ in. vacuum	Westinghouse Mfg. Co.
1	Reduction gear			Connects turbine to propeller shaft	3360 r.p.m. reduced to 90 r.p.m.	Westinghouse Mfg. Co.
1	Main condenser	Surface	4,500 sq. ft. tube surface	Attached to turbine	Produces a 28½ in. vacuum	Westinghouse Mfg. Co.
1	Pump	Turbo	400 gal. per min.	Circulating water		Westinghouse Mfg. Co.
1	Aux. condenser	Surface	800 sq. ft. tube surface	For auxiliaries		Westinghouse Mfg. Co.
2	Pumps	Centrifugal	6,000 gal. per hr.	Removal of condensate	200 lb. pressure	Westinghouse Mfg. Co.
2	Air ejectors	Cent. entrainment	Size E	Removal of air from condenser	Maintains 28½ in. vac., with 30 in. barometer and 150 lb. pressure	Westinghouse Mfg. Co.
1	Pump	Hor. duplex	6 × 5¾ × 6 in.	Lubricating oil		
1	Heater	Closed	50,000 lb. per hr.	Feedwater	Water enters at 90° F., leaves at 250° F.	Westinghouse Mfg. Co.
2	Generators	Turbo-driven	10-kw.	Electric supply	100 lb. press., 120 volts	Westinghouse Mfg. Co.
1	Refrigerating Mach.	Ammonia			Vertical engine drive	Brunswick Co.
2	Tanks		8,000 gal.	Settling oil		Merchant Ship Co.
1	Tank			Feed and filter		Merchant Ship Co.
1	Pump	Hor. simplex	6 × 5¼ × 6 in.	Sanitary pump		Worthington Pump Co.
1	Pump	Hor. duplex	10 × 12 × 12 × 12 in.	Air and circ. pump	Used in conjunction with auxiliaries	Deane Bros.
1	Pump	Hor. simplex	4½ × 3¾ × 4 in.	Fresh water		Worthington Pump Co.
1	Tank	Cylindrical	500 gal.	Light fuel oil		

Miscellaneous: 1 Inspection tank; 1 Oil cooler; 1 Oil heater; 1 Oil filter tank; 1 Oil storage tank; and 1 Oil drain tank.

The service tanks are located 150 feet from the power plant, as required by the Fire Underwriters. These tanks are 30 feet in diameter and 12 feet high, and hold 1500 barrels of oil. A 10-inch underground pipe line connects the service tanks to the Oil Company's 55,000-barrel supply tank.

The pump suction lines are carried on arms extending from pipe columns. Under each suction line a $2\frac{1}{2}$-inch exhaust line runs to the service tanks and terminates in a coil inside the tank. This coil is supplied with exhaust steam from the pumps and keeps the oil at 90 deg. fahr., at which temperature it is carried to the heaters where the temperature is raised to 240 deg. fahr. The suction lines are tied together near the service tanks and are connected by a header inside the boiler room. This permits the use of either pump on either tank as desired. From this header, four 5-inch branches are taken, two for each pump with a strainer in each branch. On top of the columns supporting the suction piping, two 2-inch high-pressure steam lines are carried to the tanks, for smothering, in case of a fire. This pipe line ends in a perforated pipe running across the top of the tank under the roof.

The front of the furnace is covered with a steel plate, Fig. 465, arranged to form the front of a blast box, which houses the registers through which the forced draft, applied by turbine-driven fans, is admitted.

The Merit automatic-control system is arranged to control the stack dampers and, by connections to the cut-out valves, the fires, in three pre-determined steps. The first step admits oil at 100 pounds pressure for low fires; the second step at 140 pounds for intermediate fires; and the third step at 160 pounds for maximum operation. These pressures give ratings varying from 50 to 250 per cent, using three or four burners.

548. Marine Steam Power Plant. — The arrangement of the equipment in a modern cargo ship of 9000 tons dead weight capacity, is shown in plan and elevation in Fig. 467. The boilers are cross-drum water-tube boilers supplying steam to a 2-cylinder turbine served by a surface condenser. Draft is furnished by an induced draft fan located on the second deck directly above the boilers. Ashes are elevated to the upper deck by an ash hoist. The type, size, and capacity of the equipment are given in Table 39.

549. Energy Distribution in a Modern Central Station Plant. — An energy distribution chart, Fig. 468, of the Colfax Steam Power Plant, shows where the greatest losses occur when developing power with steam. A careful study of this chart shows that in the modern central station the efficiency of the plant as a whole is high.

REFERENCES

Vol. 53, pages 532, 554, 806 and 841; Vol. 54, page 622; Vol. 55, pages 2, 604, 678, 716, 760 and 846; POWER.

Vol. 46, pages 397 and 447; POWER PLANT ENGINEERING.

Report of Prime Movers Committee, NATIONAL ELECTRIC LIGHT ASSOCIATION.

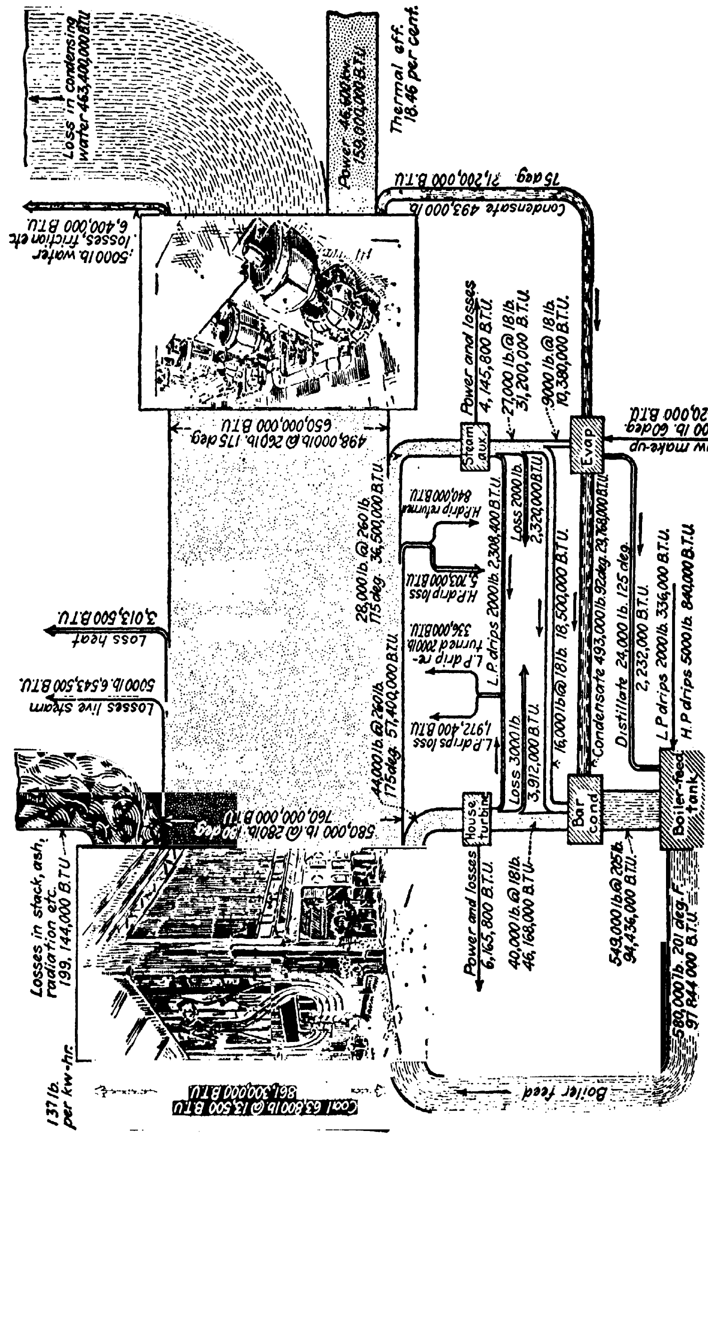


FIG. 468. — Energy Distribution in a Modern Central Station.

REVIEW QUESTIONS

1. What is the present tendency in choosing modern power plant equipment?

2. Sketch the arrangement of the equipment in the Dodge Bros. Power Plant.

3. Describe the method of handling coal at the Calumet Plant of the Chicago Edison Co.

4. Draw a chart showing the distribution of the energy in a modern central station using coal to generate steam.

5. Describe the operation of the oil-burning system installed in the power plant of the Fall River Electric Light Co.

6. What changes are necessary in a coal-fired plant, to accommodate it to a pulverized-coal-burning plant?

7. Name a common method of supplying the circulating water to the condenser.

8. What is the purpose of a house turbine?

9. Describe the arrangement of the equipment in some power plant which you have visited.

MISCELLANEOUS TABLES

TABLE 40. — AREA OF CIRCLES

Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area
1/8	0.0123	4½	15.904	16½	213.82	32	804.24	56	2463.0	80	5026.5
1/4	0.0491	5	19.635	17	226.98	33	855.30	57	2551.7	81	5153.0
3/8	0.1104	5½	23.758	17½	240.52	34	907.92	58	2642.0	82	5281.0
1/2	0.1963	6	28.274	18	254.46	35	962.11	59	2733.9	83	5410.6
5/8	0.3067	6½	33.183	18½	268.80	36	1017.8	60	2827.4	84	5541.7
3/4	0.4417	7	38.484	19	283.52	37	1075.2	61	2922.4	85	5674.5
7/8	0.6013	7½	44.178	19½	298.64	38	1134.1	62	3019.0	86	5808.8
1	0.7854	8	50.265	20	314.16	39	1194.5	63	3117.2	87	5944.6
1 1/8	0.9940	8½	56.745	20½	330.06	40	1256.6	64	3216.9	88	6082.1
1 1/4	1.227	9	63.617	21	346.36	41	1320.2	65	3318.3	89	6221.1
1 3/8	1.484	9½	70.882	21½	363.05	42	1385.4	66	3421.2	90	6361.7
1 1/2	1.767	10	78.54	22	380.13	43	1452.2	67	3525.6	91	6503.8
1 5/8	2.073	10½	86.59	22½	397.60	44	1520.5	68	3631.6	92	6647.6
1 3/4	2.405	11	95.03	23	415.47	45	1590.4	69	3739.2	93	6792.9
1 7/8	2.761	11½	103.86	23½	433.73	46	1661.9	70	3848.4	94	6939.7
2	3.141	12	113.09	24	452.39	47	1734.9	71	3959.2	95	7088.2
2 1/4	3.976	12½	122.71	24½	471.43	48	1809.5	72	4071.5	96	7238.2
2 1/2	4.908	13	132.73	25	490.87	49	1885.7	73	4185.3	97	7389.8
2 3/4	5.939	13½	143.13	26	530.93	50	1963.5	74	4300.8	98	7542.9
3	7.068	14	153.93	27	572.55	51	2042.8	75	4417.8	99	7697.7
3 1/4	8.295	14½	165.13	28	615.75	52	2123.7	76	4536.4	100	7854.0
3 1/2	9.621	15	176.71	29	660.52	53	2206.1	77	4656.0	101	8011.8
3 3/4	11.044	15½	188.69	30	706.86	54	2290.2	78	4778.3	102	8171.3
4	12.566	16	201.06	31	754.76	55	2375.8	79	4901.6	103	8332.3

TABLE 41. — PROPERTIES OF SUPERHEATED STEAM

Reproduced by permission from "Properties of Steam and Ammonia," by G. A. Goodenough. John Wiley & Sons, Inc.

Absolute Pressure Lb. per Sq. In. / Sat. Temp. °F.		Saturated Steam	Degrees of Superheat 50	100	150	200	250	300		Absolute Pressure Lb. per Sq. In. / Sat. Temp. °F.
15 (213)	H_s	1152.2	1176.2	1199.9	1244.8	1246.9	1269.3	1283.9	H_s	15 (213)
	v	26.30	28.38	30.43	32.47	34.48	36.49	38.5	v	
	ϕ	1.7573	1.7918	1.8235	1.8772	1.8806	1.9068	1.9317	ϕ	
20 (228)	H_s	1157.7	1172.3	1206.2	1226.2	1253.5	1276.9	1301.7	H_s	20 (228)
	v	21.10	21.68	23.23	24.77	26.29	27.71	29.31	v	
	ϕ	1.7343	1.7685	1.7999	1.8291	1.8562	1.8823	1.9068	ϕ	
40 (267.2)	H_s	1171.3	1196.6	1221.3	1245.6	1269.7	1293.9	1375.3	H_s	40 (267.2)
	v	10.51	11.33	12.12	12.91	13.68	14.45	14.63	v	
	ϕ	1.6788	1.7123	1.7433	1.7719	1.8789	1.8990	1.9530	ϕ	
60 (292.7)	H_s	1179.1	1205.3	1230.7	1255.5	1280.1	1283.7	1328.6	H_s	60 (292.7)
	v	7.18	7.74	8.29	8.83	9.35	9.45	9.97	v	
	ϕ	1.6462	1.6797	1.7106	1.7389	1.7654	1.7909	1.8130	ϕ	
80 (312)	H_s	1184.4	1211.3	1237.3	1262.7	1287.6	1312.4	1337.1	H_s	80 (312)
	v	5.48	5.92	6.34	6.74	7.14	7.53	7.92	v	
	ϕ	1.6227	1.6566	1.6873	1.7155	1.7417	1.7667	1.7903	ϕ	
100 (327.8)	H_s	1188.4	1215.9	1242.5	1268.3	1299.3	1318.7	1343.7	H_s	100 (327.8)
	v	4.44	4.79	5.13	5.46	5.79	6.10	6.41	v	
	ϕ	1.6045	1.6381	1.6691	1.6974	1.7234	1.7484	1.7719	ϕ	
120 (341.3)	H_s	1191.4	1219.6	1246.7	1272.9	1298.6	1303.7	1329.1	H_s	120 (341.3)
	v	3.74	4.82	5.10	5.25	5.52	5.62	5.87	v	
	ϕ	1.5893	1.6234	1.6544	1.6827	1.7092	1.7356	1.7571	ϕ	
140 (353.1)	H_s	1193.7	1217.1	1250.2	1276.8	1302.8	1328.5	1354.0	H_s	140 (353.1)
	v	3.23	3.44	3.75	3.99	4.23	4.45	4.67	v	
	ϕ	1.5762	1.6042	1.6418	1.6702	1.6964	1.7213	1.7447	ϕ	
160 (363.6)	H_s	1195.7	1225.1	1253.1	1280.1	1306.3	1332.4	1358.1	H_s	160 (363.6)
	v	2.84	3.08	3.30	3.51	3.72	3.92	4.12	v	
	ϕ	1.5649	1.5996	1.6308	1.6593	1.6856	1.7104	1.7337	ϕ	
180 (373.1)	H_s	1197.2	1227.2	1255.6	1283.1	1309.6	1335.9	1360.7	H_s	180 (373.1)
	v	2.54	2.75	2.95	3.14	3.33	3.51	3.69	v	
	ϕ	1.5547	1.5897	1.6210	1.6497	1.6759	1.7007	1.7241	ϕ	
200 (381.9)	H_s	1198.5	1229.0	1257.9	1285.7	1312.6	1339.1	1355.0	H_s	200 (381.9)
	v	2.29	2.49	2.68	2.85	3.02	3.18	3.34	v	
	ϕ	1.5456	1.5808	1.6125	1.6410	1.6676	1.6921	1.7156	ϕ	
300 (417.5)	H_s	1201.9	1235.0	1266.5	1295.4	1323.3	1350.9	1378.1	H_s	300 (417.5)
	v	1.545	1.686	1.847	1.943	2.060	2.175	2.287	v	
	ϕ	1.5092	1.5484	1.5817	1.6074	1.6343	1.6590	1.6829	ϕ	
500 (467.2)	H_s	1201.7	1238.9	1273.2	1305.5	1336.0	1365.5	1394.4	H_s	500 (467.2)
	v	0.928	1.025	1.112	1.194	1.270	1.344	1.412	v	
	ϕ	1.4601	1.4993	1.5336	1.5584	1.5919	1.6176	1.6416	ϕ	

H_s = total heat above 32 deg. fahr., B.t.u. v = specific volume, cu. ft. per lb.
ϕ = total entropy above 32 deg. fahr.

TABLE 42. — COMMON LOGARITHMS (Log_{10})

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

$e = 2.71828.$

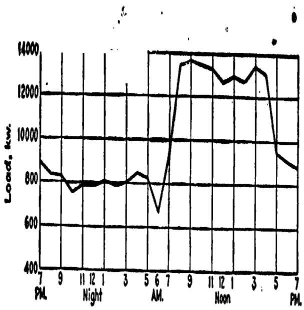
TABLE 42. — (*Continued.*) COMMON LOGARITHMS (Log_{10})

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

Naperian $\log_e = 2.302 \log_{10}$.

TABLE 43. — DECIMAL EQUIVALENTS OF FRACTIONS OF ONE INCH

$\frac{1}{64}$	.015625	$\frac{17}{64}$	.285625	$\frac{33}{64}$	.515625	$\frac{49}{64}$	.765625
$\frac{1}{32}$	.03125	$\frac{9}{32}$	.28125	$\frac{17}{32}$	.53125	$\frac{25}{32}$	.78125
$\frac{3}{64}$	.046875	$\frac{19}{64}$	.296875	$\frac{35}{64}$	.546875	$\frac{51}{64}$	.796875
$\frac{1}{16}$	.0625	$\frac{5}{16}$	.3125	$\frac{9}{16}$	.5625	$\frac{13}{16}$	.8125
$\frac{5}{64}$	.078125	$\frac{21}{64}$	.328125	$\frac{37}{64}$	.578125	$\frac{53}{64}$	.828125
$\frac{3}{32}$	.09375	$\frac{11}{32}$	.34375	$\frac{19}{32}$	.59375	$\frac{27}{32}$	.84375
$\frac{7}{64}$	.109375	$\frac{23}{64}$	.359375	$\frac{39}{64}$	.609375	$\frac{55}{64}$	.859375
$\frac{1}{8}$	.125	$\frac{3}{8}$	.375	$\frac{5}{8}$	.625	$\frac{7}{8}$	.875
$\frac{9}{64}$	.140625	$\frac{25}{64}$	.390625	$\frac{41}{64}$	.640625	$\frac{57}{64}$	.890625
$\frac{5}{32}$	.15625	$\frac{13}{32}$	.40625	$\frac{21}{32}$	.65625	$\frac{29}{32}$	.90625
$\frac{11}{64}$	.171875	$\frac{27}{64}$	.421875	$\frac{43}{64}$	.671875	$\frac{59}{64}$	.921875
$\frac{3}{16}$	.1875	$\frac{7}{16}$	.4375	$\frac{11}{16}$	.6875	$\frac{15}{16}$	.9375
$\frac{13}{64}$	.203125	$\frac{29}{64}$	.453125	$\frac{45}{64}$	.703125	$\frac{61}{64}$	.953125
$\frac{7}{32}$	.21875	$\frac{15}{32}$	.46875	$\frac{23}{32}$	.71875	$\frac{31}{32}$	.96875
$\frac{15}{64}$	.234375	$\frac{31}{64}$	.484375	$\frac{47}{64}$	.734375	$\frac{63}{64}$	.984375
$\frac{1}{4}$	.25	$\frac{1}{2}$	.50	$\frac{3}{4}$	.75	1	1.



TYPICAL ELECTRIC LOAD CURVE

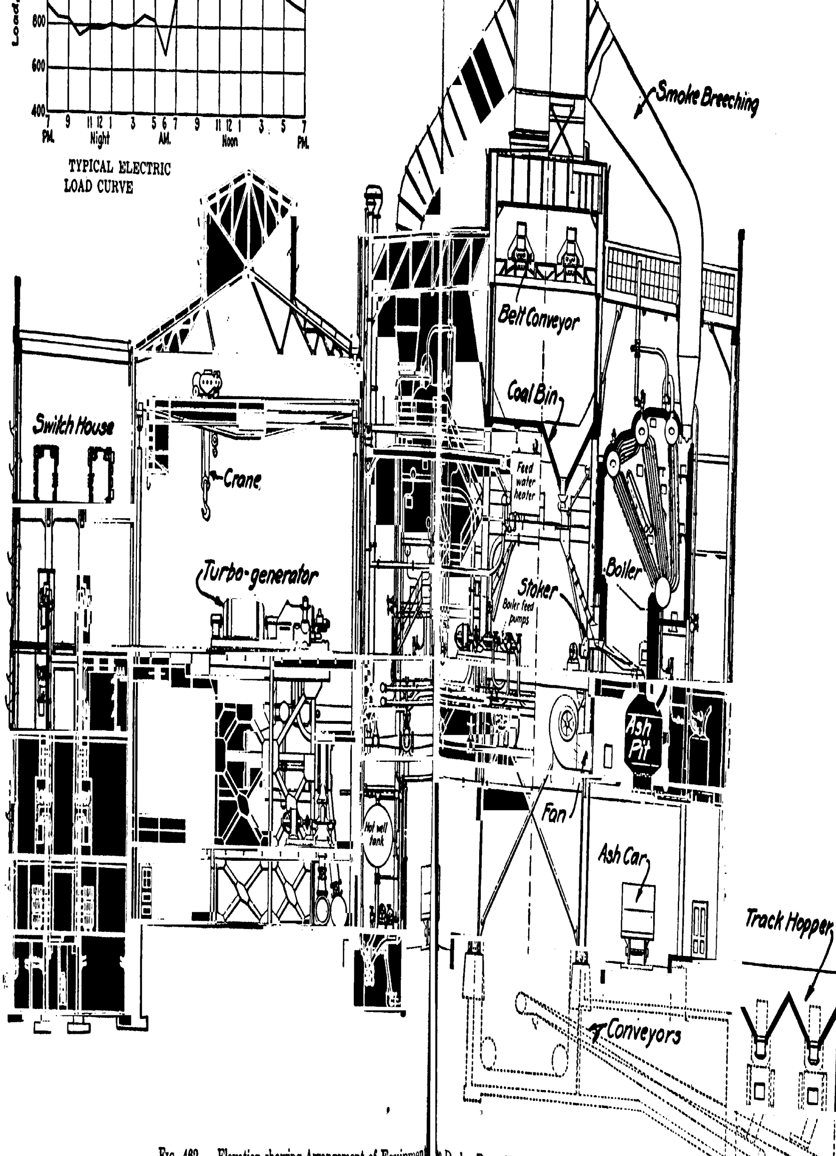


FIG. 462. — Elevation showing Arrangement of Equipment [illegible] Dodge Bros. Steam Power Plant. — Isolated Plant.

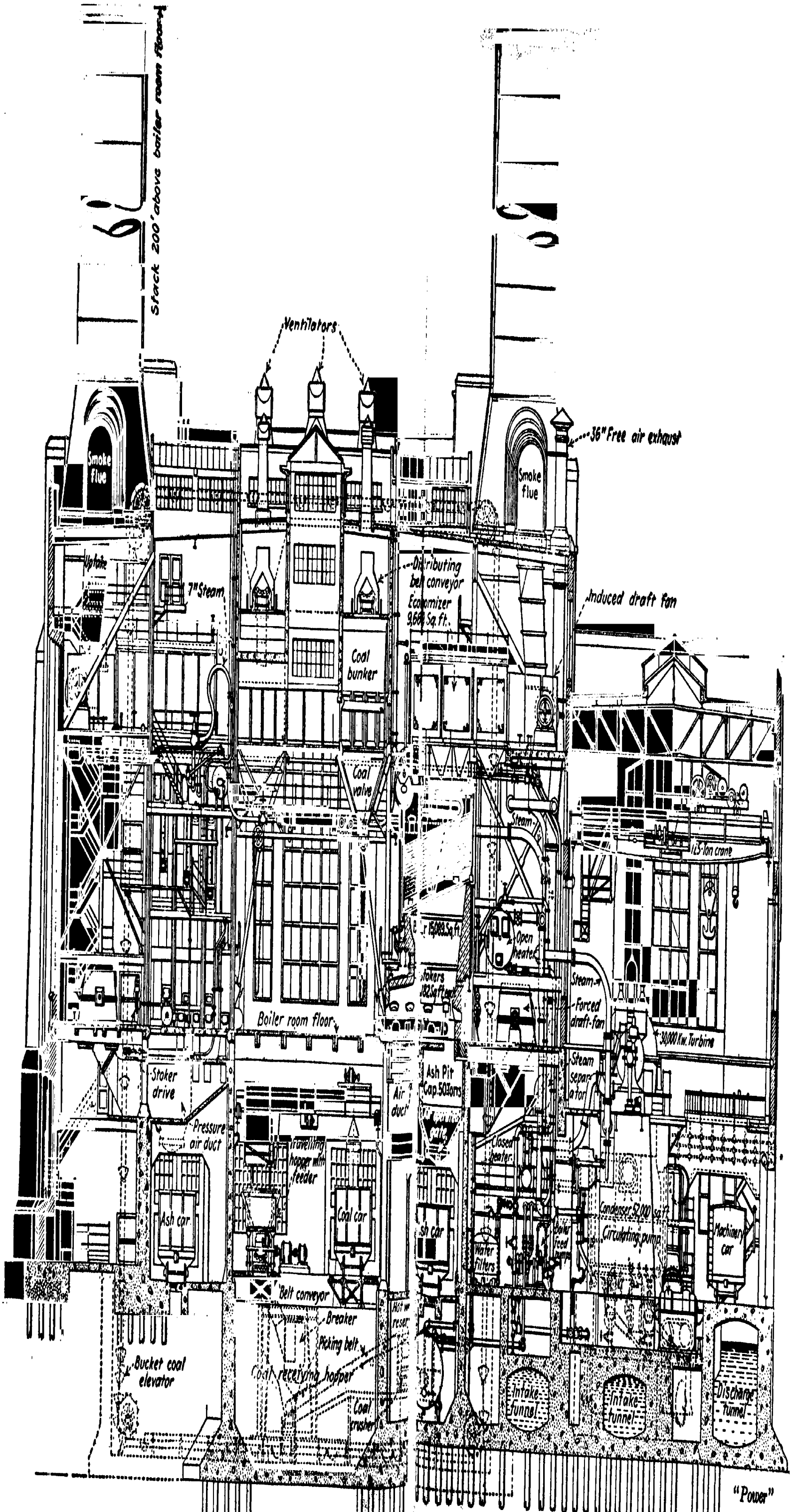


FIG. 459. — Elevation through Calumet Steam Power Plant s ing Arrangement of Equipment. — Central Station.

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